ON PROVIDING UNIFORM EDGE COMPRESSIVE LOADS FOR WIDE FLAT PLATES

Milan Louis Pittman, Jr.
and
Virgil Willard Rinehart

Letter on front cover:

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SIVE LOADS FOR WIDE FLAT PLATES

[illegible] LOUIS PITTMAN, Jr.
and
VIRGIL WILLARD RINEHART

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FOR WIDE FLAT PLATES

by

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May 1954

ABSTRACT

ON PROVIDING UNIFORM EDGE COMPRESSIVE LOADS FOR WIDE FLAT PLATES

by

Milan L. Pittman and Virgil W. Rinehart

Submitted to the Department of Naval Architecture and Marine Engineering on May 24, 1954 in partial fulfillment of the requirements for the degree of Naval Engineer.

This investigation constitutes a part of a program being carried out in the Department of Naval Architecture and Marine Engineering under the auspices of the Society of Naval Architects and Marine Engineers.

Many merchant vessels are built using a transverse system of framing. These transverse frames, and the more widely spaced longitudinal stiffeners divide the ship's plating into short wide rectangular panels, which are placed either in tension or compression across the short dimension as the ship "hogs" or "sags" under the effects of loading and wave motion. In general the compressive loading is more critical due to the phenomenon of "buckling".

For test purposes these panels can be represented by wide flat plates restrained to varying degrees at their edges and loaded in compression across the short dimension. In this investigation an apparatus was designed for the determination of the buckling and ultimate loads of such plates when all four edges were simply supported (no rotational restraint). This apparatus is intended to accommodate sixteen groups of plates having all possible combinations of the following length-width ratios (a/b) and length-thickness ratios (a/t): a/b = 1/4, 1/3, 1/2, and 3/4; a/t = 30, 50, 70, 90. Only one of these

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For test purposes these panels can be represented by flat plates restrained to varying degrees at their edges and loaded in compression across the short dimension. A test apparatus was designed for the determination of the buckling and ultimate loads of such plates when all four edges were simply supported (no rotational restraint). This apparatus is intended to accommodate sixteen groups of plates having all possible combinations of the following length-width ratios (a/b) and length-thickness ratios (a/t): $a/b = 1/4$, $1/3$, $1/2$, and $3/4$; $a/t = 30$, 40, 50, 60. This report covers

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groups was actually tested, the remainder being left for future completion of the program.

From the work carried out the following conclusions are drawn:

(1) While further work is necessary to make the apparatus completely satisfactory, the equipment designed and built provided reasonably uniform distribution of the compressive load, and a very close approximation of simple support on all four edges of a test plate 56 3/4" in width, 13 5/16" in length, and 5/32" in thickness.

(2) A group of four plates was tested in buckling to augment the evaluation of the test apparatus. From these tests the following tentative conclusions are drawn:

(a) The uniformity of stress distribution is not as critical as was anticipated.

(b) The strength of the plate is not a function of the measured maximum initial unfairness alone, the initial contour and the past history being important also.

(c) For the four plates tested ($a/b = 0.237$; $a/t = 86.4$) the experimental critical stresses obtained supported the prediction for the given ratios according to Bleich [8].

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(c) For the four plates tested ($a/b = 0.362$; $a/t = 85.2$) the experimental critical stresses obtained supported the prediction for the given ratios according to Bleich [8].

Cambridge, Massachusetts
May 24, 1954

Professor Leicester F Hamilton
Secretary of the Faculty
Massachusetts Institute of Technology
Cambridge 39, Massachusetts

Dear Sir:

In accordance with the requirements of the degree of Naval Engineer, we herewith submit a thesis entitled "On Providing Uniform Edge Compressive Loads for Wide Flat Plates".

ACKNOWLEDGMENT

The authors wish to express their deep appreciation of the patient and helpful guidance of Professor C. H. Norris throughout the whole period of this endeavor.

ON PROVIDING UNIFORM EDGE COMPRESSIVE

LOADS FOR WIDE FLAT PLATES

Table of Contents

NOTATION

a	- Unsupported length between centers of loading segments
b	- Unsupported length between "a"-edge supports
b'	- Actual width of plate
d	- Lateral deflection of center of plate from original plane
d_o	- Initial lateral deflection of center of plate (zero load)
d'	- Distance from point on horizontal center line of plate to reference line
k	- Coefficient in buckling formula (based on b/t)
$\bar{k}$	- " " " " (based on a/t)
n	- Number of half waves in buckled plate
t	- Thickness of plate
w	- Lateral deflection of an arbitrary point on the neutral plane of the plate from the original plane
x	- Distance in direction of load (origin at horizontal cl.)
y	- " normal to " (origin at one edge)
A	- Edge area of plate - b't
C	- Arbitrary constant in solution of differential equation
D	- Plate constant - $AI/(1-\nu^2)$
E	- Young's modulus of elasticity
E_t	- Tangent modulus
F	- Friction force in one jack
I	- Moment of inertia of strip of plating 1" wide, t thick - $t^3/12$
L	- Same as "a" - used in column formulae
P	- Applied load
P'	- Load transmitted to test plate
P_{cr}	- Critical load

P_{ult} - Ultimate load

P_i - Internal pressure in jack manifold

S - Force exerted by jack plunger return spring

α - a/b - aspect ratio or width ratio

$\varepsilon_{1,2}$ - Measured strains for pair of opposite gages

ε_a - Longitudinal strain at neutral axis of plate

ε_{av} - Average longitudinal strain - P/AE

f - Coefficient of edge restraint ($f = \infty$ for simple support)

μ - Micro- (one-millionth)

ν - Poisson's ratio - assumed to be 0.3

σ - Stress intensity

σ_a - Longitudinal stress intensity at neutral axis

σ_{av} - Average longitudinal stress intensity - P/A

σ_{cr} - Critical stress intensity

σ_{ult} - Ultimate stress intensity

σ_y - Yield stress

τ - E_t/E

I. INTRODUCTION

A. General

This investigation represents a part of a more extensive program being sponsored by the Hull Structures Committee of the Society of Naval Architects and Marine Engineers. (See Appendix A.) This project is designed to determine the buckling and ultimate strengths of flat plates of such dimensions and boundary conditions as those usually found in transversely framed merchant vessels.

The framing of a vessel may be considered, for the purposes of such an investigation, as the boundaries of the various panels into which it divides the ship's skin or plating. Each of these panels may be acted upon by direct axial loads, shear forces, and/or moments along either or both edges, and in addition by pressure forces normal to the plane of the plate. Due to the behavior of the plating as a whole, the edges of a particular panel may be considered as simply supported, elastically restrained, or in the extreme condition, clamped. Variables which affect the plating strength are: (1) the ratio of plating length to thickness; (2) the ratio of length to width; (3) the degree of restraint against rotation of the plate edges; (4) the stiffness of panel boundaries, (5) effect of non-uniform distribution of the load.

B. Research Project

It was felt by the Hull Structures Committee that the experimental data on the strength of these "wide" plates was incomplete and that a considerable range of plating sizes and shapes utilized in ship construction had not been investigated at all. As a specific point, Montgomerie's

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B. Research Project

It was felt by the Hull Structures Committee that the experimental data on the strength of these "thin" plates was incomplete and that a considerable range of plating sizes and shapes utilized in ship construction had not been investigated at all. As a specific point, investigations results [36], based upon tests on plates with unloaded edges free and loaded edges elastically restrained have been applied to conditions where they are not valid, in lieu of better data. Accordingly a research project was formulated which had as its aims the accomplishment of the following four purposes:

Phase I To make a survey of the literature relating to the theoretical buckling and ultimate strengths and experimental work thereon; to determine the ranges of parameters for which experimental data was lacking or insufficient; and to find information upon the effect and growth of unfairness.

Phase II To conduct buckling and ultimate strength tests for uniaxial edge compression on plates of such dimensions as are indicated by Phase I, with loaded and unloaded edges simply supported.

Phase III To conduct buckling and ultimate strength tests for uniaxial edge compression on plates of such dimensions as are indicated by Phase I, with unloaded edges simply supported and loaded edges elastically restrained to varying degrees.

Phase IV To design, build, and evaluate a test apparatus for the accomplishment of Phases II and III.

It was the original purpose of the authors to complete Phases I, II, and IV. However, due to delays in delivery of the test equipment, and to difficulties encountered in getting the test equipment to function as desired, Phase IV was not completed to the entire satisfaction of the authors. One group of plates under Phase II was tested but its results

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C. Literature Survey

A search of the literature on, or pertaining to, buckling ultimate strength of flat plates under various loadings and edge conditions was made to determine whether adequate theory and experimental data existed pertinent to the range of plate parameters and edge conditions under investigation.

The results of this search led to the belief that theory presented by F. B. Bleich [8] was an adequate, though not mathematically precise, basis for judging the proposed test results, both for buckling and ultimate strength in the elastic and plastic ranges.

The search failed to reveal sufficient experimental data in the region of interest. Shultz and Weschler [45] ran five tests at $a/b = 1/2$ with a/t varying from 20 to 120 with all edges simply supported. Yoshiki 59 also presents similar data skirting the range of interest, with the closest data at $a/b = 1/3$ and $1/2$ but $a/t > 108$. Figure 1A is provided to indicate the range of interest, the proposed test parameters, and the parameters for which data is presently available.

There is also some possibility that other data found may be reevaluated into usable form when considering elastic support of the loaded edges. Various shapes were used as edge restraints but no attempt was made to evaluate the degree of restraint provided.

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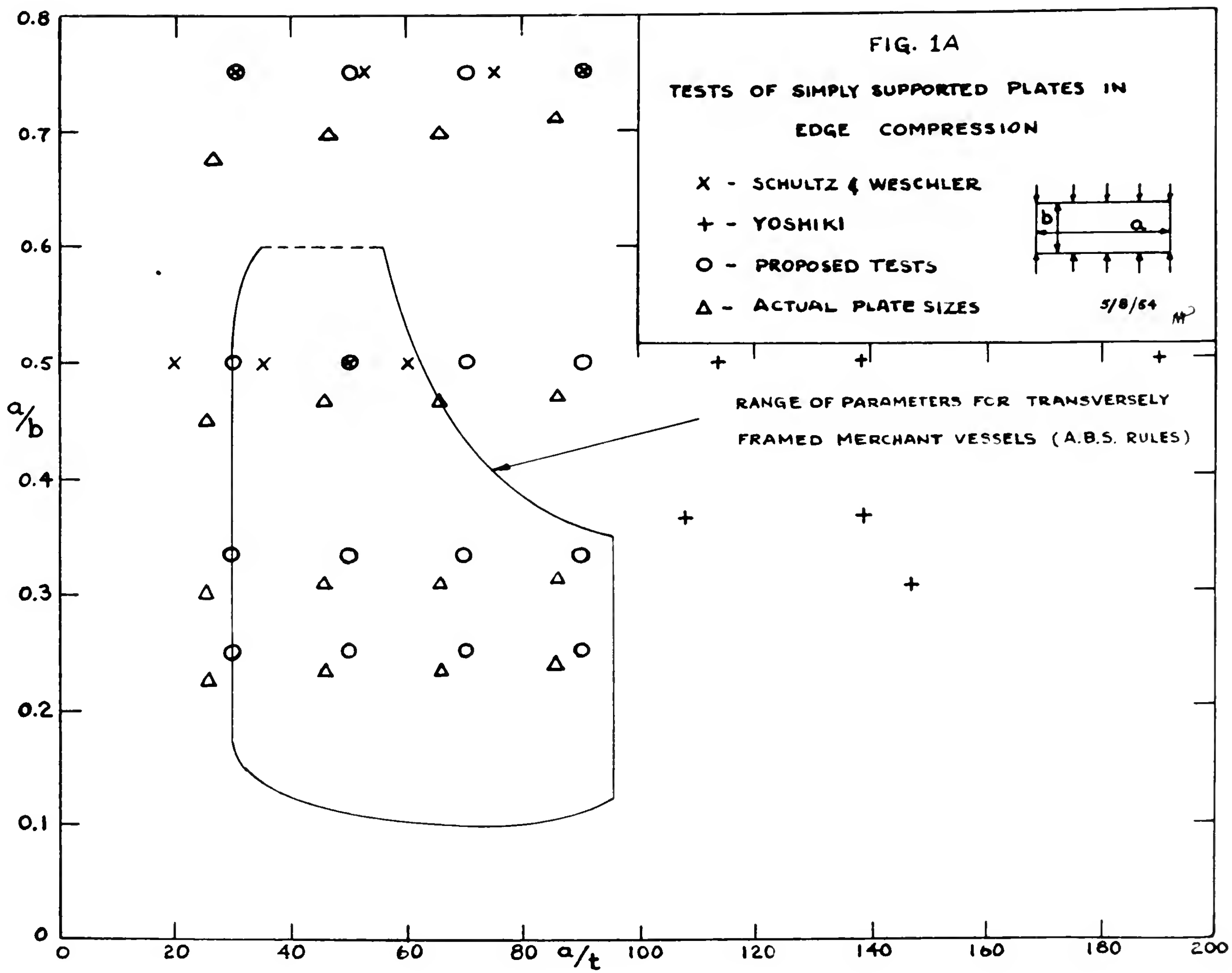
FIG. 1A
TESTS OF SIMPLY SUPPORTED PLATES IN EDGE COMPRESSION
X - SCHULTZ & WESCHLER
+ - YOSHIKI
O - PROPOSED TESTS
Δ - ACTUAL PLATE SIZES
5/8/64
b
a
RANGE OF PARAMETERS FOR TRANSVERSELY FRAMED MERCHANT VESSELS (A.B.S. RULES)
a/b
a/t
0
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
20
40
60
80
100
120
140
160
180
200

D. Experimental Program

After considering the region of interest and the points covered by previous investigators it was decided to use sixteen different size test plates having combinations of the following ratios:

a/b = 1/4 , 1/3 , 1/2 , 3/4

a/t = 30 , 50 , 70 , 90

These particular values were chosen for the following reasons:

1. Within the region of interest there has been very little data obtained, and the theory has not been confirmed.

2. Some of the a/b values lie above the region outlined by the American Bureau of Shipping Regulation, since the ABS values are for plates near the midship section, and the plates farther forward or aft have higher a/b ratios.

3. For extremely low a/b ratios, repeatable results are difficult to obtain, and it was decided to concentrate on getting usable results for the intermediate values.

4. Higher a/b ratios are of interest primarily for longitudinally framed ships.

Actual sizes for the test specimens were decided upon after consideration of the following factors:

1. The a/b, a/t ratios decided upon above.

2. In order to use existing equipment previously used in investigations by Polychrone [40] and Capozzoli [12] , the sum of one of each short dimension chosen was not to exceed 34".

3. The long dimensions were calculated from the short dimensions and the a/b ratios selected above.

4. The maximum plate thickness which could be accomodated by the

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1. The a/b, a/t ratios decided upon above.

2. In order to use existing equipment previously used in investigations by Polychrone [10] and Tapscott [illegible], the [illegible] of each short dimension chosen was not to exceed 30".

3. The long dimensions were calculated from the short dimensions and the a/b ratios selected above.

4. The maximum plate thickness which could be accommodated by the

"a-edge" supports was 5/16". In addition it was desirable to use standard plate thicknesses.

5. Since it was desired to obtain both buckling and ultimate strengths, the plates should fail under a load less than the capacity of the testing machine (300,000 pounds).

6. The "a" and "b" dimensions, as determined above were then altered as follows:

(a) The "a" dimension was decreased by 3/4" since it was believed at this time that the loading segments which transmitted the load would act as a part of the effective length of plate. (It was later decided that this was not the case, and hence the actual a/b and a/t values were not exactly the same as had been selected.) The actual plate dimensions are given in Table I.

(b) The "b" dimension was increased by 1/2" to allow 1/4" extension beyond the line of ball bearing support at each unloaded edge.

TABLE I

Plating Dimensions

Nominal a/b		Nominal a/t 30	50	70	90
3/4	l	6.750	10.188	10.188	13.313
	w	10.500	15.080	15.080	19.25
	t	1/4	7/32	5/32	5/32
	a/b	.675	.699	.699	.710
	a/t	27.0	46.5	65.2	85.2
1/2	l	6.750	10.188	10.188	13.313
	w	15.500	22.375	22.375	28.625
	t	1/4	7/32	5/32	5/32
	a/b	.450	.465	.465	.474
	a/t	27.0	46.5	65.2	85.2
1/3	l	6.750	10.188	10.188	13.313
	w	23.000	33.513	33.513	42.688
	t	1/4	7/32	5/32	5/32
	a/b	.300	.310	.310	.316
	a/t	27.0	46.5	65.2	85.2
1/4	l	6.750	10.188	10.188	13.313
	w	30.500	44.25	44.25	56.75
	t	1/4	7/32	5/32	5/32
	a/b	.225	.233	.233	.237
	a/t	27.0	46.5	65.2	85.2

II. PROCEDURE

A. General

The analysis of the behavior of flat plates is based upon a given stress distribution, in this case the distribution of the force applied at the loaded edges. The problem, then, was to provide a means of applying a uniformly distributed edge load and at the same time to maintain simple support at the loaded edges, so that the edges of the plate were allowed complete freedom of rotation and freedom of translation in the plane of the plate while being restricted from any lateral movement.

B. Load Distribution

A uniform load distribution can be obtained by one of two methods, either by causing a uniform contraction in the plane of the plate, or by applying a load which is distributed by some external system. Either of these methods provides the desired loading condition prior to buckling of a perfectly flat plate. In the case of a plate in a ship's structure, the assumption that the "loaded" edges of a panel remain straight (uniform deflection loading) is probably closer to the actual situation than is the assumption of a uniform load distribution. However for reasons given below, this method of loading is very difficult to use.

The available testing machine, shown in Fig. 34, has an upper loading head which is a hollow cylindrical forging approximately 18" outside diameter, and 11" inside; the lower head is 24" square. Since the largest plate is 56 3/4" wide, some auxiliary device was necessary to distribute the load to the plate. Preliminary calculations indicated that an I-beam

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The available testing machine, shown in Fig. 4, has an upper loading head which is a hollow cylindrical forging approximately 12" outside diameter, and 11" inside; the lower head is 24" square. Since the largest plate is 56 3/4" wide, some auxiliary device was necessary to distribute the load to the plate. Preliminary calculations indicated that a beam which would limit the diminution of the stress near the ends of the plate to a nominal value, say 5% of the average, would be of prohibitive size. This analysis is borne out by the fact that the constant deflection heads of the testing machine at the David Taylor Model Basin are extremely deep.

The solution commensurate with the time and money available was the use of a system of seven 20-ton Blackhawk Porto-Power hydraulic jacks connected to a common manifold, identical systems being used at both the top and bottom of the plate. The testing machine transmitted the load through an 18-inch wide flange I-beam to the jacks which in turn applied the load to the plate through the arrangement intended to provide simple support. This system, while not exactly reproducing the practical situation, provided a solution which was feasible, and gave known conditions of loading.

Bridging the gaps between the jacks was a 3/4" x 3-5/8" x 5' bar of AISI C-1045 steel hardened to 50 Rockwell C. The purpose of this bar was to further distribute the load between the jacks when they were widely separated, and served as a bearing surface against which the loading segments bore. The bar was ground flat on one side, and on the other had a shallow groove ground to allow two alternative ways of using the bar.

The loading segments transmitted the load from the loading bar to the plate itself. These segments were cut from 3/4" diameter drill rod which had a longitudinal groove milled to center depth; the rod was then cut to lengths of 7/8" and 1/2", the shorter segments being placed at the end of the plate. These segments were hardened to a hardness comparable to that of the loading bar. These segments transmitted the load along a radius to the edge of the plate. In this way torsional restraint along which would limit the distribution of the stress near the ends of the plate to a nominal value, say 5% of the average, would be of prohibitive size. This analysis is borne out by the fact that the constant deflection beams of the testing machine at the David Taylor Model Basin are extremely deep.

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Bridging the gaps between the jacks was a 1/2" x 3-5/8" x 9' bar of AISI C-1045 steel hardened to 50 Rockwell C. The purpose of this bar was to further distribute the load between the jacks when they were widely separated, and served as a bearing surface against which the loading segments bore. The bar was ground flat on one side, and on the other had a shallow groove ground to allow two alternative ways of using the bar.

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In the design of the segments, sufficient clearance was allowed in the milled slot so as to reduce the probability of a notch failure at the internal corners. To reduce the amount of eccentricity of loading, shimming strips 0.017 inches thick were inserted between the plate sides and the segments. (See Figure 27.)

After the first test showed that the load distribution was quite erratic, a gasket of neoprene was inserted between the plate edges and the loading segments for the purpose of eliminating local hard spots. In later tests gaskets of 3/32" and double 1/16" diameter rosin core solder were used for this same purpose. Although the problem of hard spots was not completely solved by this means, it was reduced so as to indicate the use of the gaskets was beneficial.

C. Instrumentation

To obtain the required strain measurements, Type A-3 SR-4 electric strain gages were mounted on opposite sides of the plate at the center of the plate, and in some cases at additional locations along the horizontal center line. For the purpose of determining the distribution of the applied load, gages were also mounted along the top and bottom edges as close to the edge as possible to measure the least favorable distribution. (See Fig. 1B.) A four-arm Wheatstone bridge having as active gages, the gage on the plate and a compensating dummy gage on an unloaded plate of the same thickness, was used; the strains were read on a Baldwin the plate edge was negligible, the plate edge was free to rotate, and the sliding friction and/or the groove in the loading bar prevented lateral movement of the plate edges. It is believed that a very close approximation to simple support was achieved.

In the design of the segments, sufficient clearance was allowed in the milled slot so as to reduce the probability of a notch failure at the internal corners. To reduce the effect of nonuniformity of loading, aluminum strips 0.017 inches thick were inserted between the plate sides and the segments. (See Figure 17.)

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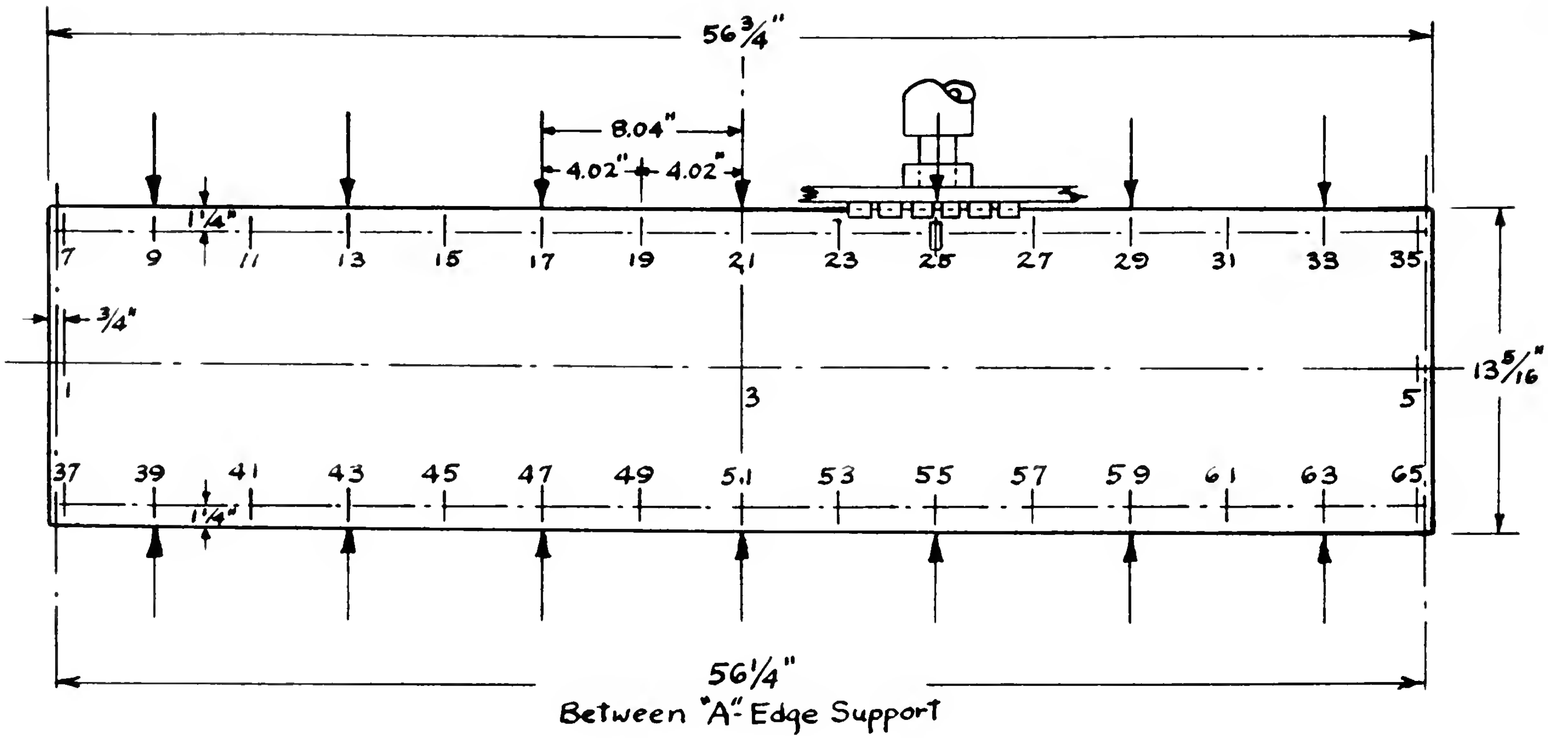
FIG.
TEST PLATE
No. 1



Details of Locations of Jacks & Strain Gages
Note: Even numbered gages on opposite side

Type K (or L) Strain Indicator. Strain readings could be read to ± 2 microinches/inch. Although the accepted accuracy is ± 5 microinches/inch, differences of ± 10 microinches/inch between initial and final zero readings were not uncommon, and occasional shifts of greater magnitude were recorded. However, in most cases the zero shift was from 0-8 microinches/inch.

Lateral deflections were measured by an Ames dial gage with a range of 0.500 inch, mounted so as to slide along a planed angle iron at the horizontal center line of the plate. (See Figs. 30 and 32.) The dial gage itself was accurate to within ± 0.001 inch. However, even though the angle upon which the gage was mounted was calibrated when the system was unloaded, no attempt was made to check whether or not it deformed any when the system was loaded.

The total load applied was measured on Massachusetts Institute of Technology testing machine No. 105, a hydraulic installation built by Southwark-Emery with a capacity of 300,000 pounds. On the lower scale (0-30,000 pounds) loads could be read to ± 10 pounds, and when the machine was used to apply the load, the load could be maintained to ± 50 pounds of the desired value. When the hand pump was used to apply the load, comparable accuracy could be obtained, the load readings being taken on the testing machine.

As an additional check on the total load, a Babcock and Wilcox gage with a range of 0-1000 psi was used to measure the pressure at the manifold of the upper jack system for some tests.

D. Preparation of Test Plates

The test plates were provided by Bethlehem Steel Co., Shipbuilding

Type K (or L) Strain Indicator. Strain readings could be read to ± 2 microinches/inch. Although the accepted accuracy is ± 5 microinches/inch, differences of ± 10 microinches/inch between initial and final zero readings were not uncommon, and occasional shifts of greater magnitude were recorded. However, in most cases the zero shift was from 0-5 microinches/inch.

Lateral deflections were measured by an Ames dial gage with a range of 0.500 inch, mounted so as to slide along a planed angle iron at the horizontal center line of the plate. (See Figs. 30 and 32.) The dial gage itself was accurate to within ± 0.001 inch. However, even though the angle upon which the gage was mounted was calibrated when the system was unloaded, no attempt was made to check whether or not it deformed any when the system was loaded.

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As an additional check on the total load, a Babcock and Wilcox gage with a range of 0-1000 psi was used to measure the pressure at the manifold of the upper jack system for some tests.

D. Preparation of Test Plates

The test plates were provided by Bethlehem Steel Co., Shipbuilding

Division, Quincy, Mass. They were laid out in the same direction and cut from a single steel plate. Of the sizes of plates for which the apparatus was designed it is considered that the group of plates with the lowest aspect ratio (1/4), i.e., those having the longest loaded edge, presented the most difficult problem in securing a reasonably uniform load distribution. Hence this group of four plates was selected to evaluate the apparatus. The opposite edges of all test plates were planed parallel to each other.

In order to mount strain gages on the plates it was found expedient to sand-blast or shot-blast these plates in order to remove the mill scale. One of this group of plates, Plate No. 1, was selected and instrumented along both edges as shown in Fig. 1B. SR-4 type A-3 strain gages were mounted close to each loaded edge so as to fall under and in between each of the jacks. Gages were mounted in opposite pairs on both sides of the plate and connected so each gage could be read individually. Hence both bending moments and direct stress can be found. Plate No. 2 was instrumented with five pairs of gages located on the center line of the plates along the "b" dimension as shown in Fig. 23. Plates Nos. 3 & 4 each have one pair of gages located at the center of the plate.

Due to the shearing, machining, and blasting, the test plates were slightly curved in both directions. This curvature was minimized as far as practicable by subsequent rolling. It was found that the curvature remaining in the long direction could be straightened by inserting blocking between the plate and the angles connecting the "a"-edge supports. These blocks were removed before the first readings were taken. When the grooved side of the loading bar was used, this was not necessary. The curvature along the shorter dimension is treated as an initial unfairness.

Division, Quincy, Mass. They were laid out in the same direction and cut from a single steel plate. Of the sizes of plates for which the apparatus was designed it is considered that the group of plates with the lowest aspect ratio (L/b), i.e., those having the longest loaded edge, presented the most difficult problem in securing a reasonably uniform load distribution. Hence this group of four plates was selected to evaluate the apparatus. The opposite edges of all test plates were planed parallel to each other.

In order to mount strain gages on the plates it was found expedient to sand-blast or shot-blast these plates in order to remove the mill scale. One of this group of plates, Plate No. 1, was selected and instrumented along both edges as shown in Fig. 12. SR-4 type A-1 strain gages were mounted close to each loaded edge so as to fall under and in between each of the jaws. Gages were mounted in opposite pairs on both sides of the plate and connected so each gage could be read individually. Hence both bending moments and direct stress can be found. Plate No. 2 was instrumented with five pairs of gages located on the center line of the plates along the "b" dimension as shown in Fig. 13. Plates Nos. 3 & 4 each have one pair of gages located on the center of the plate.

Due to the shearing, machining, and blasting, the test plates were slightly curved in both directions. This curvature was minimized as far as practicable by subsequent rolling. It was found that the curvature remaining in the long direction could be straightened by inserting blocking between the plate and the angles connecting the "b"-edge supports. These blocks were removed before the first readings were taken. When the grooved side of the loading bar was used, this was not necessary. The curvature along the shorter dimension is treated as an initial deflection.

During the first run on Plate No. I (Fig. 18) the specimen was inadvertently buckled and an ultimate load of 35000 pounds was recorded. At this time it was expected that this ultimate load would be higher. Hence it was necessary to re-straighten Plate No. I before further testing could proceed. This was accomplished successfully using the testing machine as a press.

E. Evaluation of Test Apparatus

A total of eighteen tests on Plate No. I were run with the test apparatus to determine the load distribution. In each of these tests some detail of the loading system was changed, such as the jacks (with or without), loading bar, gasket, shimming, or loading segments. In all of these tests the same plate was used since a total of 60 gages were installed on it for this purpose and it was not practical to instrument more plates this fully. In each test a load of about 20,000 pounds was applied and held for several minutes to set the gasket. The load was then removed, and re-applied in progressive steps, gages being read at several values of the load.

F. Buckling Tests

Buckling tests were made on the plate used in part D above, and on three other plates of the same size. It should be pointed out that these tests were made to augment the evaluation of the machine and not primarily to obtain buckling data.

In each of the buckling tests, the load was applied by means of a hand pump connected to the manifold of the upper jack system. Zero readings of the strain gages were taken when the plates were in place

During the first run on plane No. [illegible] (Fig. [illegible]) the operator was inadvertently unskilled and an extensive area of [illegible] ground was [illegible]. At this time it was observed that the estimate [illegible] would be higher. Hence it was necessary to [illegible] [illegible] the cracking would proceed. This was accomplished successfully using the heating machine as a guide.

E. Evaluation of Test Apparatus

A total of eighteen tests on [illegible] were run with the test apparatus to determine the load distribution. In each of these tests some detail of the loading system was changed, such as the grade of adhesive, loading bar, gaskets, calibrating or loading sequences. In all of these cases the same plate [illegible] were fastened on it for this purpose [illegible] instrument [illegible]. At each [illegible] pounds was applied and held for several minutes to set the [illegible]. The load was then removed, and [illegible], [illegible] read at several values [illegible].

F. Cracking Tests

Cracking tests were run on the plates used [illegible] as well as three other plates of the same size. It should be pointed out that these tests were made to compare the [illegible] and [illegible] primarily to obtain cracking data.

At each of the cracking tests, the load was applied by hand [illegible] were connected to the manifold of the [illegible] readings of the strain gages were made [illegible] ready for testing but before any load was applied. The load was applied in increments of 2000 pounds and strain gage readings taken. At certain loads further readings were taken for gages other than those giving buckling data in order to check the load distribution. Some lateral deflections of the plate were also recorded.

To obtain the ultimate load, the lower head of the test machine was moved up until the load began to decrease. This was taken as the ultimate load.

ready for testing but before any load was applied. The load was applied in increments of 2000 pounds and strain gage readings taken. At certain loads further readings were taken for gages other than those giving buckling data in order to check the load distribution. Some lateral deflections of the plate were also recorded.

To obtain the ultimate load, the lower head of the test machine was moved up until the load began to decrease. This was taken as the ultimate load.

III. RESULTS

A. Equipment Evaluation Tests

1. General

The results of evaluation tests run on test plate No. 1 (see Fig. 1B) are presented graphically in Figures 1 to 20 inclusive. These plots depict average strain readings 1 1/4" from each loaded edge of the plate under and between each hydraulic jack position along the "b" edges. Original data and sample calculations on selected tests are given in Appendices G and D respectively.

To supplement the abbreviated description of the test on each of the Figures 1 to 20 inclusive there is presented here a code of test description. A more detailed description will be found in Appendix B.

2. Code of Test Description

a. Load Application

(1.) Machine - Load applied with 300,000 # hydraulic testing machine, No. 105, Testing Materials Laboratory.

(2.) Pump - Load applied with hydraulic hand pump through top system of jacks.

b. Jacks

(1.) Seven 20-ton hydraulic jacks on each edge as shown in Fig 27 and connected to a common manifold. (Fig. 30).

c. No Jacks

(1.) All jacks removed. Loading bar bearing directly on I-beams.

III. RESULTS

A. Equipment Evaluation Tests

1. General

The results of evaluation tests run on test plate No. 1 (see Fig. 13) are presented graphically in Figures 1 to 20 inclusive. These plots depict average strain readings 1 1/2" from each loaded edge of the plate [illegible] between each hydraulic jack position along the "b" edges. Original data and sample calculations on selected tests are given in Appendices C and D respectively.

To supplement the abbreviated description of the test on each of the Figures 1 to 20 inclusive there is presented here a code of test description. A more detailed description will be found in Appendix B.

2. Code of Test Description

a. Load Application

(1.) Machine - Load applied with [illegible] universal testing machine, No. [illegible], Testing Materials [illegible].

(2.) Pump - Load applied with hydraulic hand pump through [illegible] system of jacks.

b. Jacks

(1.) Seven 50-ton hydraulic jacks on each edge as shown in Fig 27 and connected to a common manifold. (Fig. 30)

c. No Jacks

(1.) All jacks removed. Loading was applied directly on I-beams.

d. Bar - Loading Bar

(1.) 3/4" - Designates hardened steel bar 3/4" x 3 5/8" See Fig. 27.

(a) (F) - Flat side of bar toward loading segments.

(b) (G) - Grooved side of bar toward loading segments.

(2.) 1/16" - Designates thickness of cold rolled steel bar - 1 1/2" wide.

(3.) 1/4", 5/16", 3/8" - Designates thickness of cold, rolled steel bar 1 1/4" wide.

e. Gasket - Description of gasket material between edge of test plate and loading segments

(1.) Neoprene - 1/16" x 1/8" cut from neoprene sheet.

(2.) Solder - strip cut from roll of rosin core solder.

a) 3/32" - Diameter. Single strip per edge of plate

b) 1/16" - Diameter. Double strip side by side per edge of plate

f. Shims

(1.) 0.017" shim stock inserted between each side of segment groove and side of test plate. Remaining clearance $\approx$ 0.003"

g. No Shims

(1.) Shims removed. Clearance between segment slot and plate approximately 0.035".

h. Load

(1.) Gage reading of 300,000 # testing machine.

d. Bar - Loading bar

(1.) 3/4" - Designates hardened steel bar 3/4" x 3 5/8"

See Fig. 27.

(a) (7) - Flat side of bar toward loading segments.

(b) (0) - Grooved side of bar toward loading segments.

(2.) 1/16" - Designates thickness of cold rolled steel bar - 1 1/2" wide.

(3.) 1/8", 5/16", 3/8" - Designates thickness of cold rolled steel bar 1 1/4" wide.

e. Gasket - Description of gasket material between edge of test plate and loading segments

(1.) Neoprene - 1/16" x 1/4" cut from neoprene sheet.

(2.) Solder - strip cut from roll of rosin core solder.

a) 3/32" - Diameter. Single strip per edge of plate

b) 1/16" - Diameter. Double strip side by side per edge of plate

f. Shims

(1.) 0.007" thin stock inserted between each side of segment groove and side of test plate. Resulting clearance ≅ 0.002"

g. No Shims

(1.) Shims removed. Clearance between segment slot and plate approximately 0.015".

h. Load

(1.) Gage reading of 300,000 # testing machine.

i. Test Plate Orientation

(1.) Orientation indicated by showing location by number of end gages on the test plate.

j. Computed Strain

(1.) Average computed strain for a load of 10,000 lbs., 40.3 micro inches per inch, is shown as a horizontal dashed line.

B. <u>Buckling Tests</u>

Four plate specimens were tested to ultimate failure in the testing apparatus. Results of the tests are shown in Table II and Fig. 21.

Deflections and strain gage readings are shown in Appendix H.

The failure of each plate was observed to be one noded in both a and b directions.

In each test the load was applied with the hand pump and shims were inserted between the plate and leveling segments. Plates 1 & 2 were tested with double 1/16" diameter solder gaskets and with the flat side of the 3/4" loading bar toward the segments. Plates 3 & 4 were tested with single 3/32" diameter solder gasket with the grooved side of the 3/4" bar towards the loading segments.

2. Test Plate Orientation

(1.) Orientation indicated by showing location by number of and gages on the test plate.

3. Computed Strain

(1.) Average computed strain for a load of 10,000 lbs., 10.3 micro inches per inch, is shown as a horizontal dashed line.

B. <u>Buckling Tests</u>

Four plate specimens were tested to ultimate failure in the testing apparatus. Results of the tests are shown in Table II and Fig. 21. Deflections and strain gage readings are shown in Appendix B. The failure of each plate was observed to be one noted in both a and b directions.

In each test the load was applied with the head ramp and shims were inserted between the plate and leveling supports. Plates 1 & 2 were tested with double 1/16" diameter solder gaskets and with the flat side of the 3/4" locking bar toward the supports. Plates 3 & 4 were tested with single 3/32" diameter solder gasket with the grooved side of the 3/4" bar towards the loading supports.

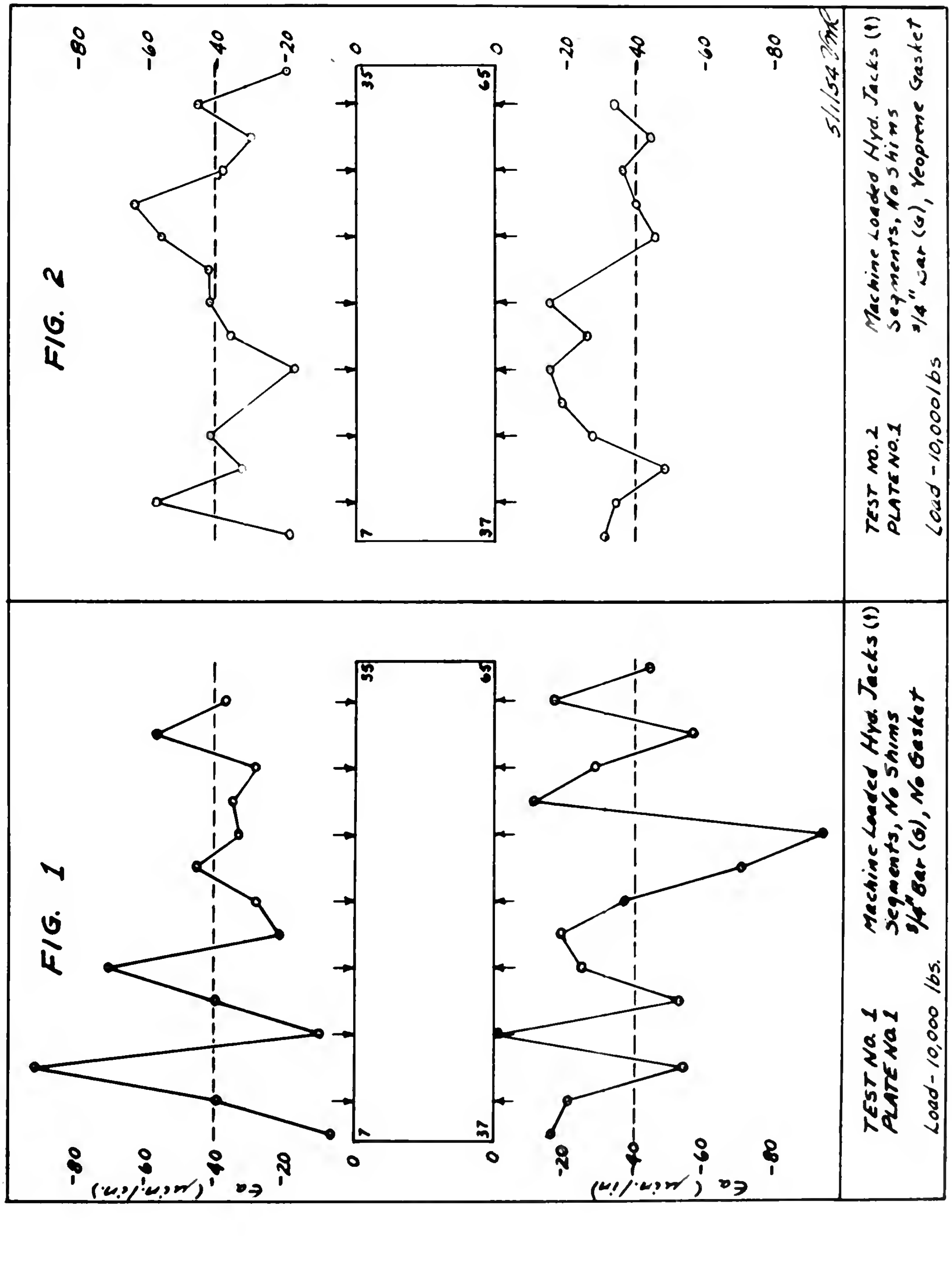
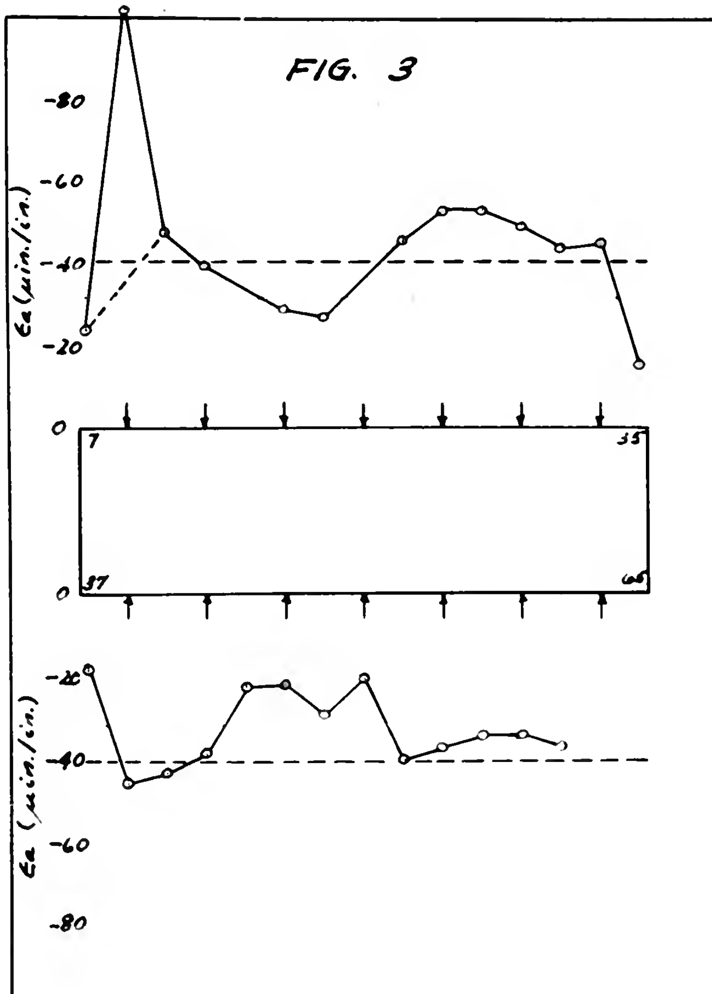


FIG. 1
FIG. 2
εa (μin./in.)
-80
-60
-40
-20
0
7
35
37
65
TEST NO. 1
PLATE NO. 1
Load - 10,000 lbs.
Machine Loaded Hyd. Jacks (4)
Segments, No Shims
3/4" Bar (G), No Gasket
TEST NO. 2
PLATE NO. 1
Load - 10,000 lbs
Machine Loaded Hyd. Jacks (4)
Segments, No Shims
3/4" Bar (G), Neoprene Gasket
5/1/54

FIG. 3

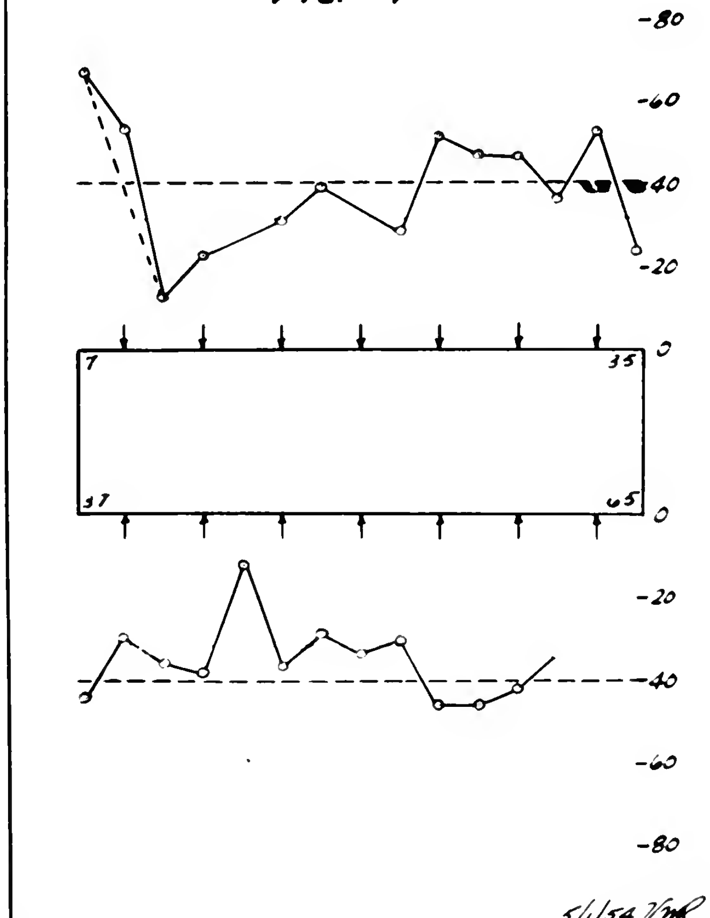


TEST NO. 3
PLATE NO. 1

Load- 10,000 lbs.

Machine Loaded Jacks (4)
Segments, No shims
3/4" Bar (G), 3/32" Solder gasket

FIG. 4

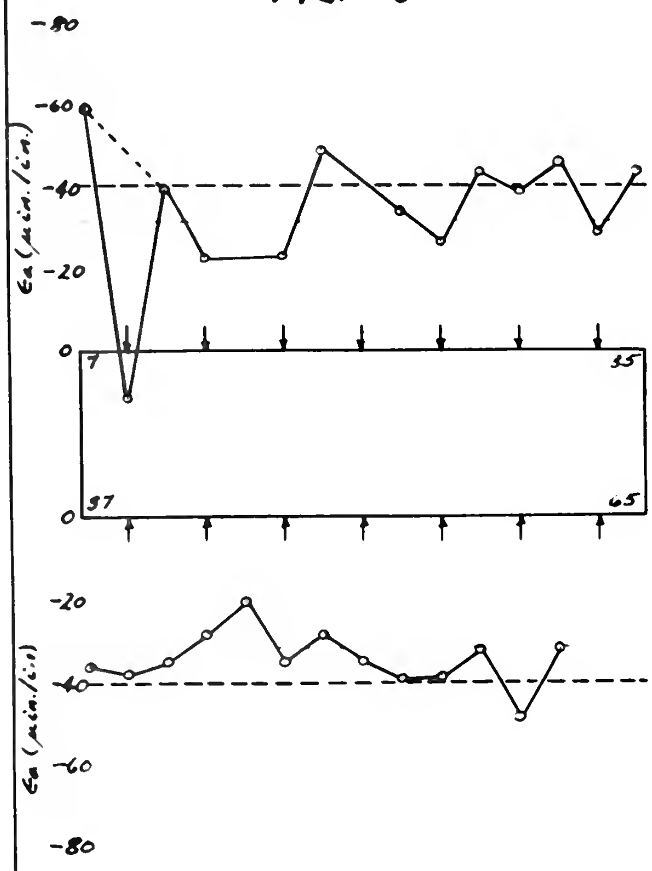


TEST NO. 4
PLATE NO. 1

Load - 10,000 lbs.

Machine Loaded Hyd. Jacks
Segments, No Shims
3/4" Bar (F), Re-used Gasket From Test No. -

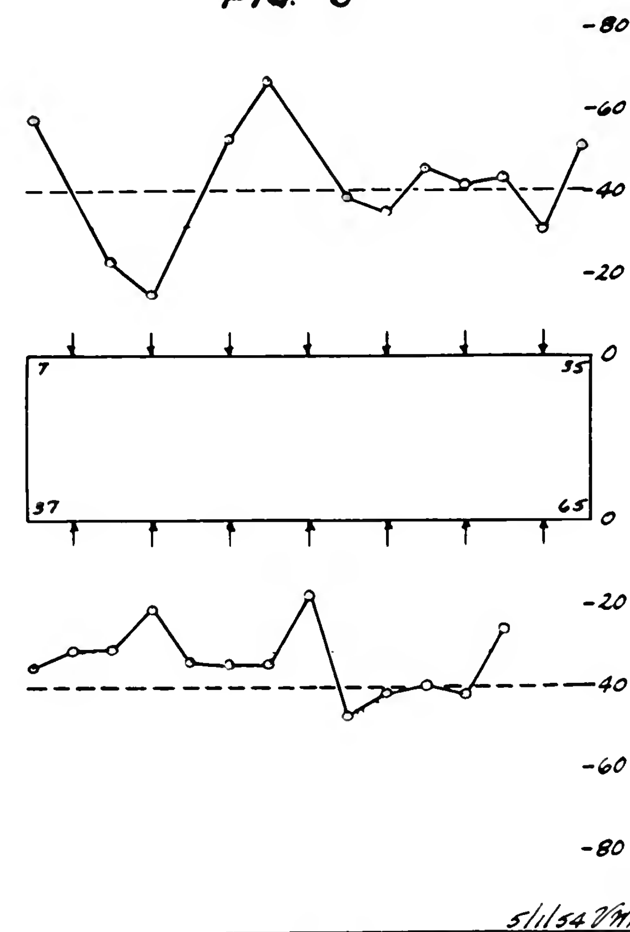
FIG. 5



TEST NO. 5
PLATE NO. 1
Load - 10,000 lbs.

REPEATABILITY TEST
SAME Conditions as Test No. 4
except for new gasket

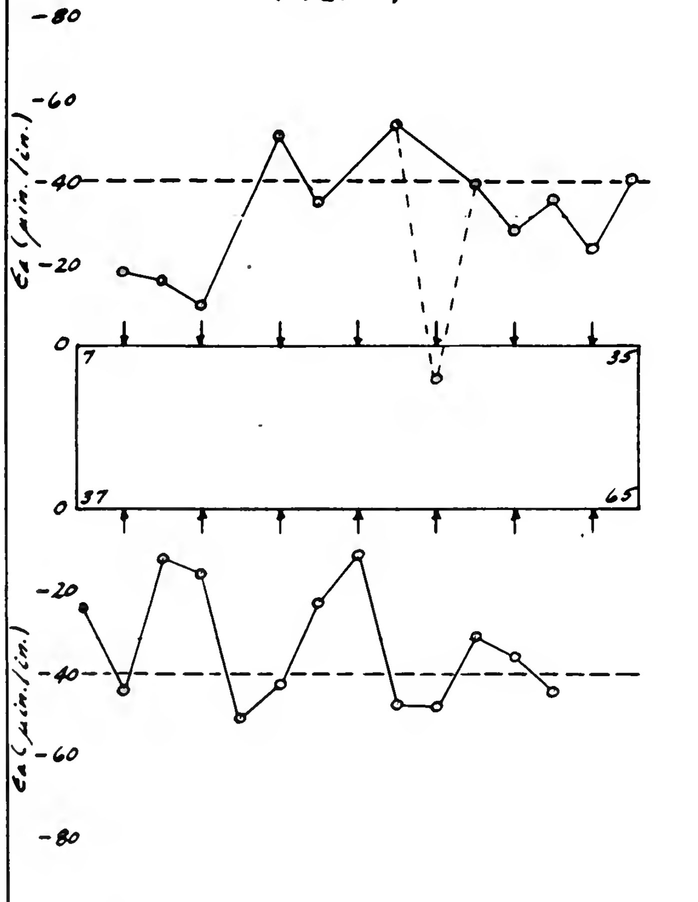
FIG. 6



TEST NO. 6
PLATE NO. 1
Load - 10,000 lbs.

Repeatability Test
Same Conditions as Nos. 4 & 5
Same gasket as Test No. 5

FIG. 7

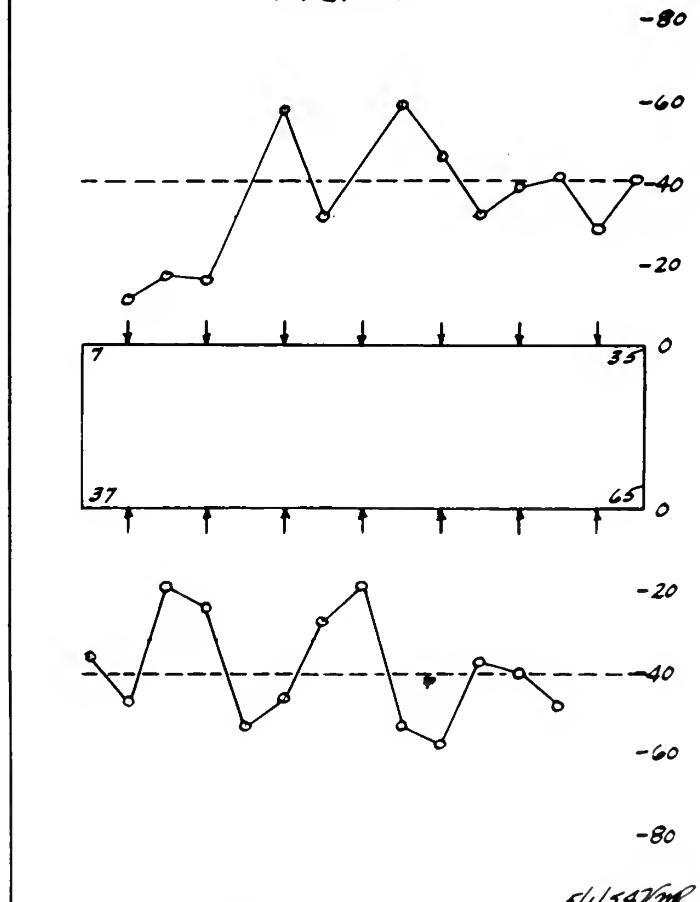


TEST NO. 7
PLATE NO. 1

Load- 10,000 lbs

Machine Loaded Hyd. Jacks
No Segments, No Shims
3/4" Bar (F), Sheet Lead Gasket ($\frac{1}{16}$") - 2 in. wide

FIG. 8

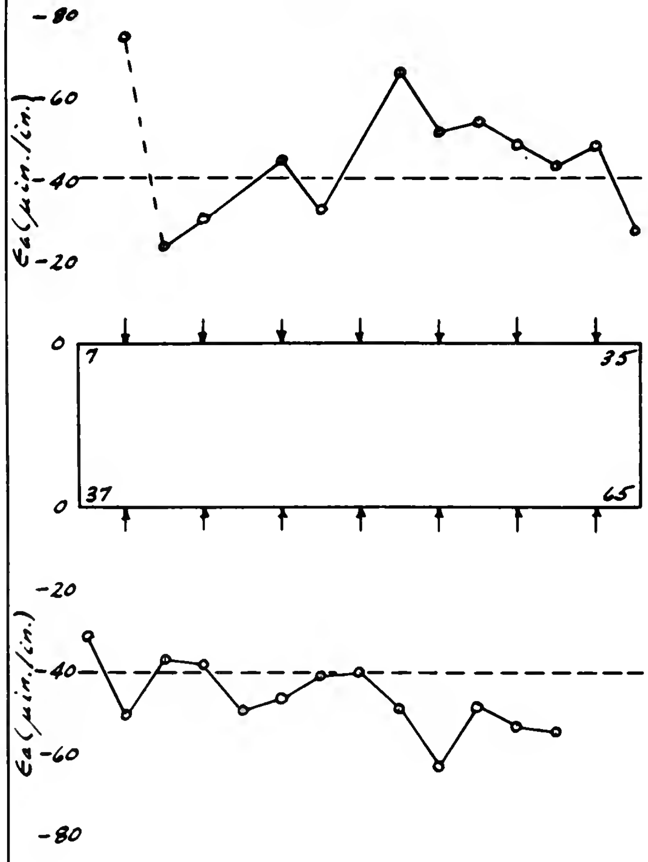


TEST NO. 8
PLATE NO. 1

Load - 10,000 lbs

Repeatability Test
Same Conditions as Test No. 7
except for new gasket
1/16" - 1 1/4" wide

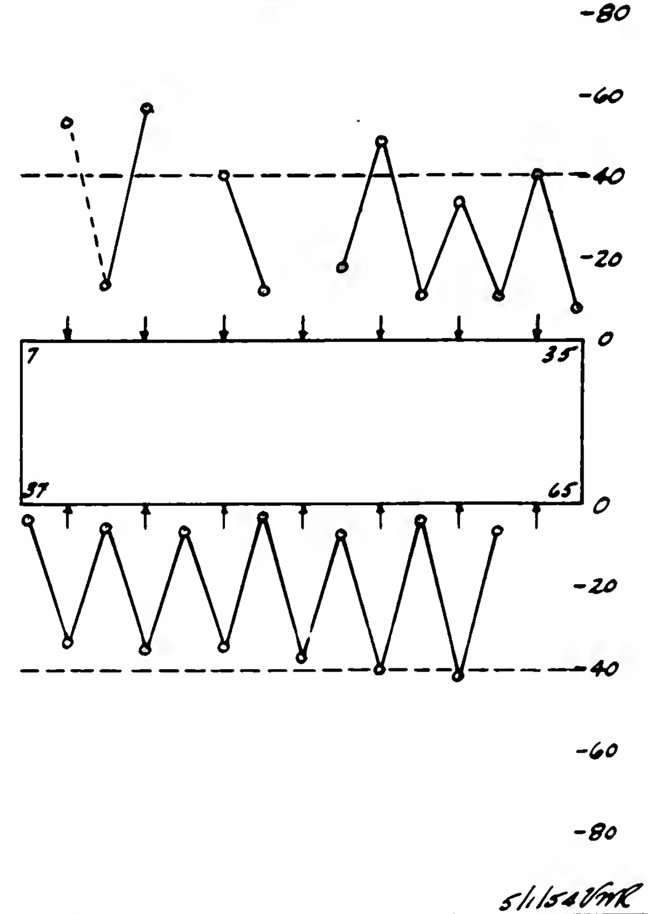
FIG. 9



TEST NO. 9
PLATE NO. 1
Load - 10,000 lbs.

Machine Loaded Hyd. Jacks
Segments, Shims
3/4" Bar (F), 3/32" Solder Gasket

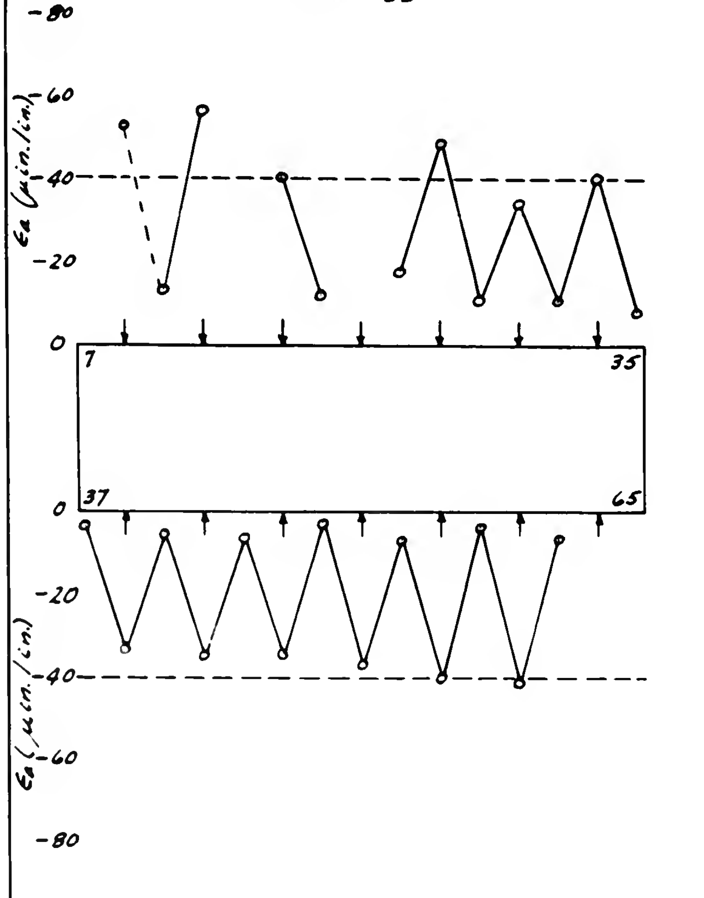
FIG. 10



TEST NO. 10
PLATE NO. 1
Load - 10,000 lbs.

Same Conditions as Test No. 9
except for replacement of
3/4" Bar with 1/16" × Strip

FIG. 11

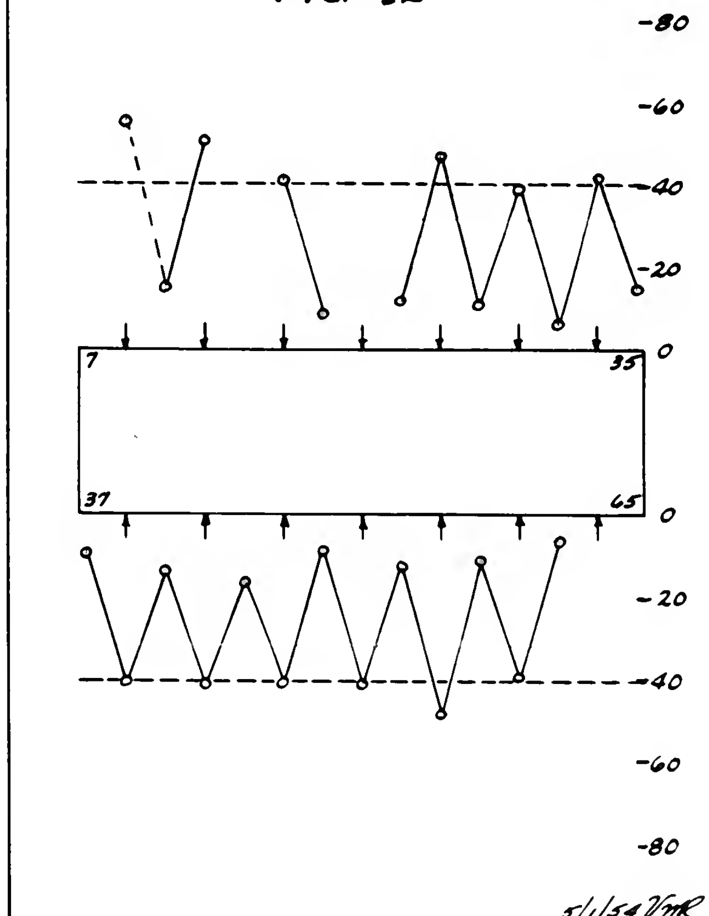


TEST NO. 10
PLATE NO. 1

Load - 10,000 lbs.

Machine Loaded Hyd. Jacks (?)
Segments, Shims
1/16" Strip; 3/32" Solder Gasket

FIG. 12

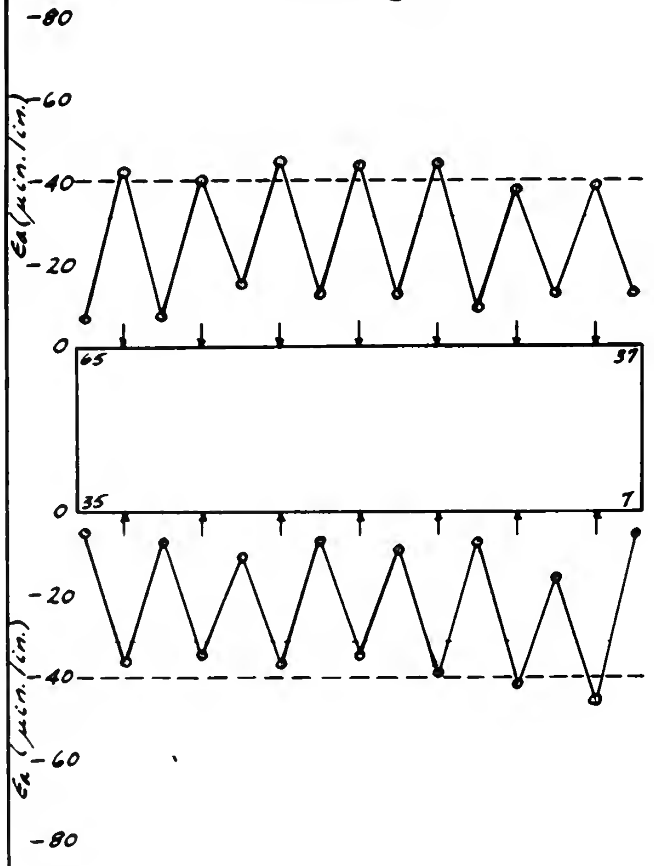


TEST NO. 11
PLATE NO. 1

Load - 10,000 lbs.

Conditions same as for
Test No. 10, except load
applied by hand pump.

FIG. 13

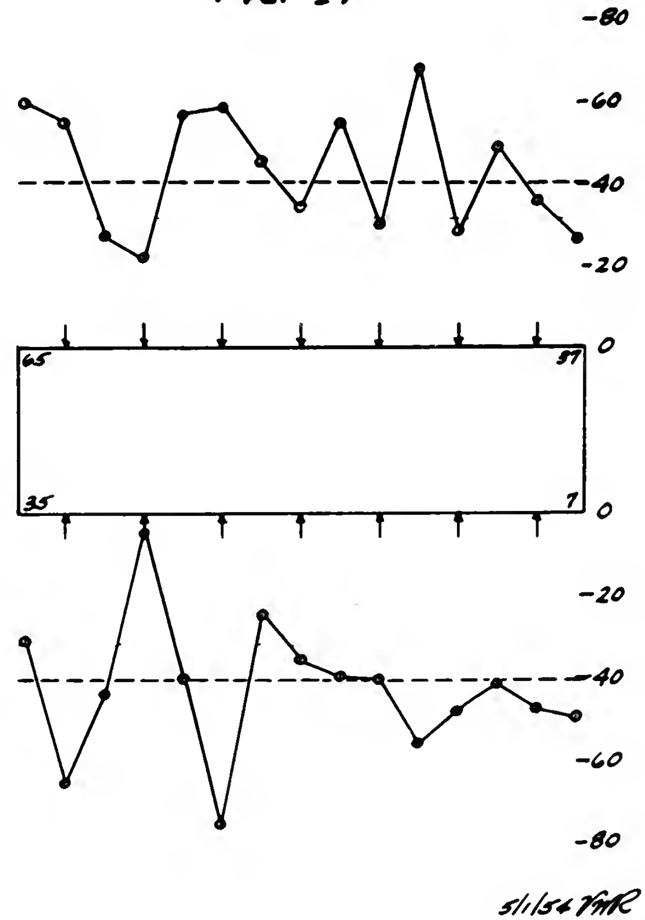


TEST NO. 12
PLATE NO. 1

Load - 10,000 lbs.

Pump Loaded Hyd. Jacks (4)
Segments, Shims
1/16" Strip, Double 1/16" Solder Gasket

FIG. 14

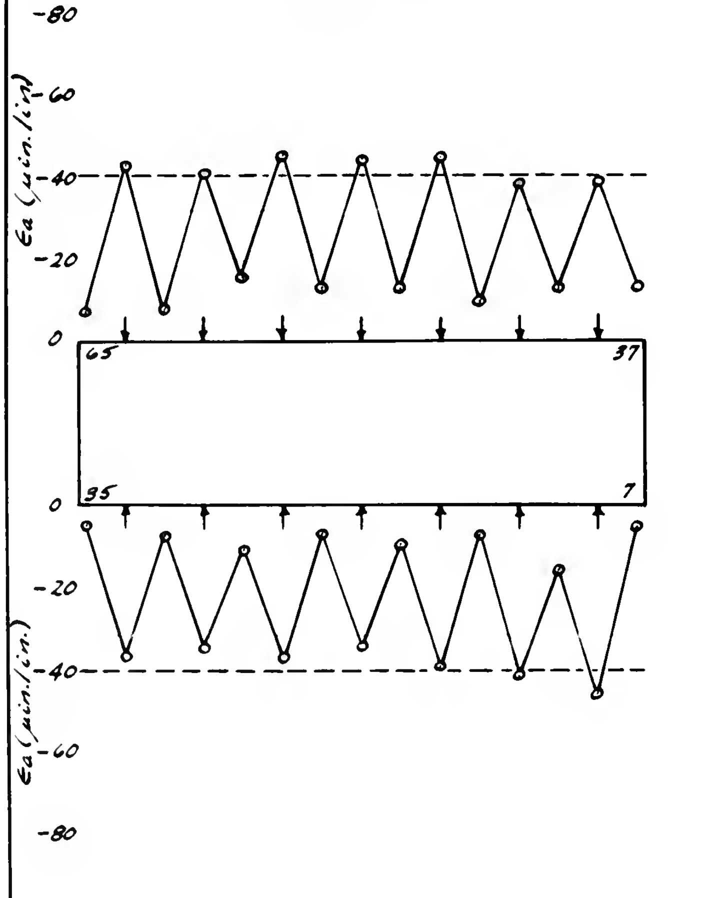


TEST NO. 13
PLATE NO. 1

Load - 10,000 lbs.

Pump Loaded Hyd. Jacks (4)
Segments, Shims
3/4" Bar (F) Re-used Gasket from Test No. 12

FIG. 15

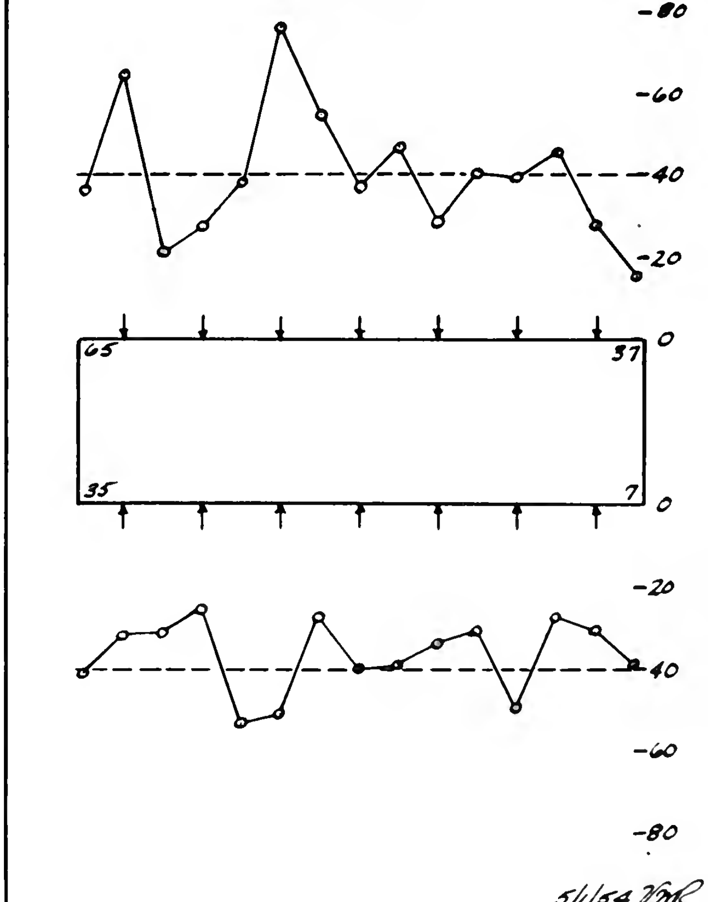


TEST NO.12
PLATE NO.1

Load- 10,000 lbs.

Pump Loaded Hyd. Jacks(↑)
Segments, Shims
1/16" Strip, Double 1/16" Solder Gasket

FIG. 16

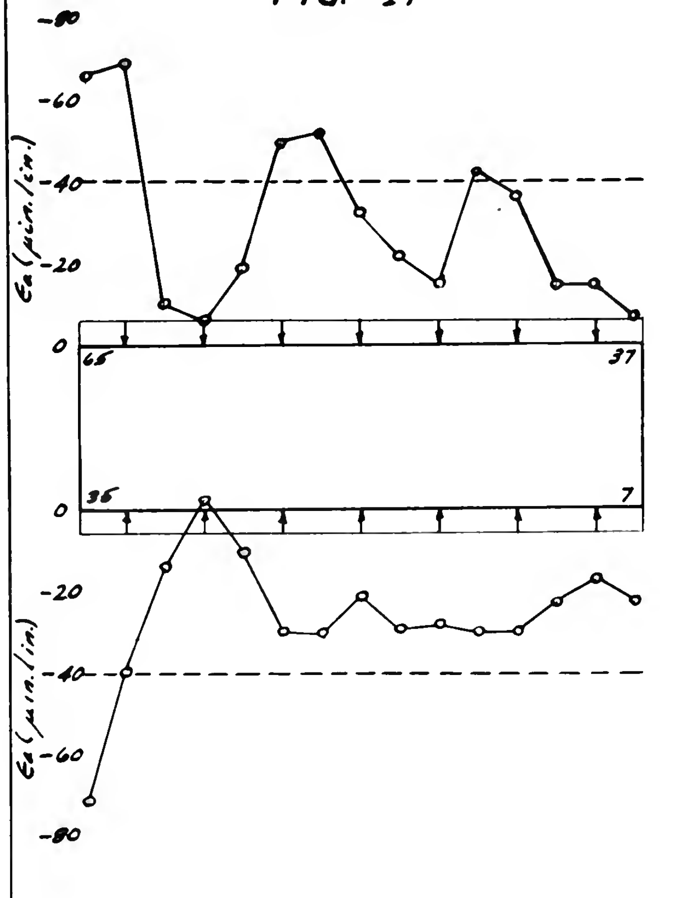


TEST NO. 14
PLATE NO. 1

Load-10,000 lbs.

Pump Loaded Hyd. Jacks (↑)
Segments, Shims
3/4" Bar (F), New Double 1/16" Solder Gasket

FIG. 17

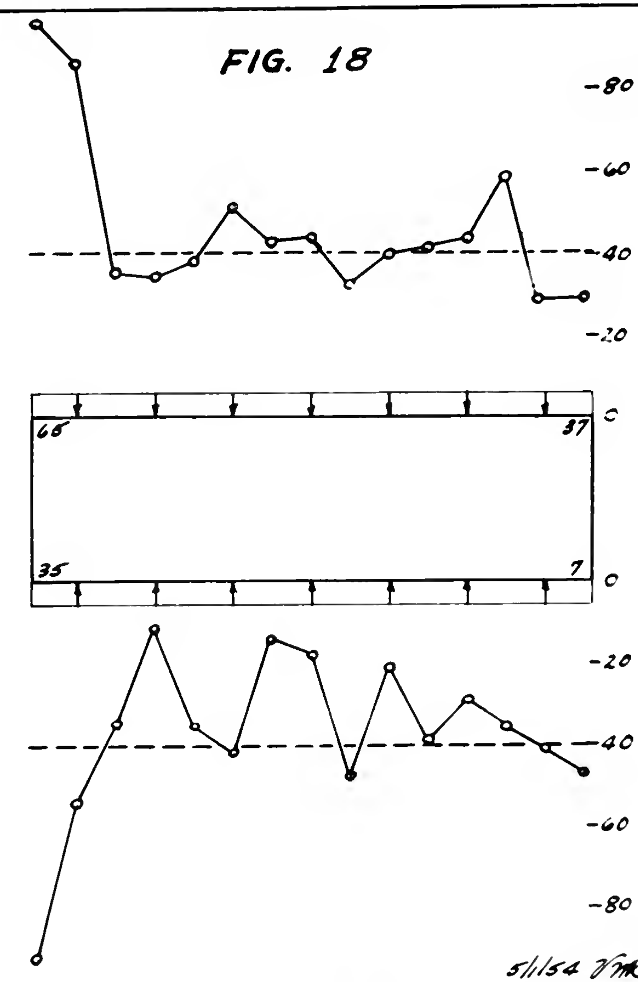


TEST NO. 15
PLATE NO. 1

Load - 10,000 lbs.

No Jacks
Segments, Shims
3/4" Bar (F), Re-used Gasket from Test No. 14

FIG. 18



TEST NO. 16
PLATE NO. 1

Load - 10,000 lbs.

Repeatability Run. Conditions same as for Test No. 15, except for new double solder (1/16") gasket.

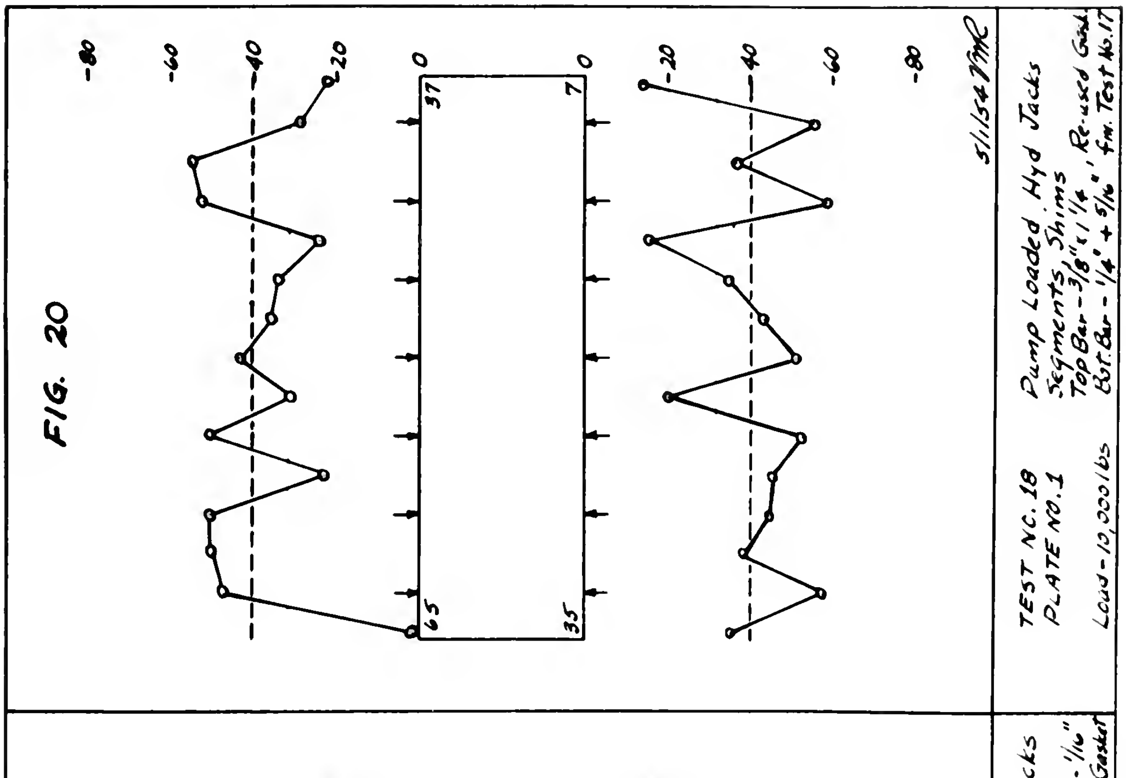


FIG. 19
FIG. 20
-80
-60
-40
-20
0
-20
-40
-60
-80
εa (μ in./in.)
65
37
35
7
TEST NO. 17
PLATE NO. 1
Load - 10,000 lbs
Pump Loaded Hyd. Jacks
Segments, Shims
Top Bar - 5/16" x 1 1/4", New 2-1/16"
Bot. Bar - 1/4" x 1 1/4" Solder Gasket
TEST NO. 18
PLATE NO. 1
Load - 10,000 lbs
Pump Loaded Hyd Jacks
Segments, Shims
Top Bar - 3/8" x 1 1/4", Re-used Gask.
Bot. Bar - 1/4" + 5/16" fm. Test No. 17
5/1/54

TABLE II

RESULTS of BUCKLING TESTS

Plate Data: a/b 0.237

a/t 86.5 (av)

E 28,300 kips/in.^{2}

Material: Mild steel (1020 ship plate)

Boundary Conditions: Simple support on all four edges.

Plate No.	Critical Strain Gage Location from Plate Center*	Initial Deflection (inches)	t (inches)	Ultimate Load (Kips)
1.	8.04" left 1.32" up	0.085	0.1535	35.0 36.2 **
2.	10.06" left on center line	0.057	0.1545	29.5
3.	at center	0.009	0.1545	32.5
4.	at center	0.071	0.1540	33.5

* Pair of gages on opposite sides of the plate and nearest the maximum deflection of plate under load. Long axis of plate is horizontal.

** Ultimate load after re-straightening and after many stress cycles to loads below the ultimate.

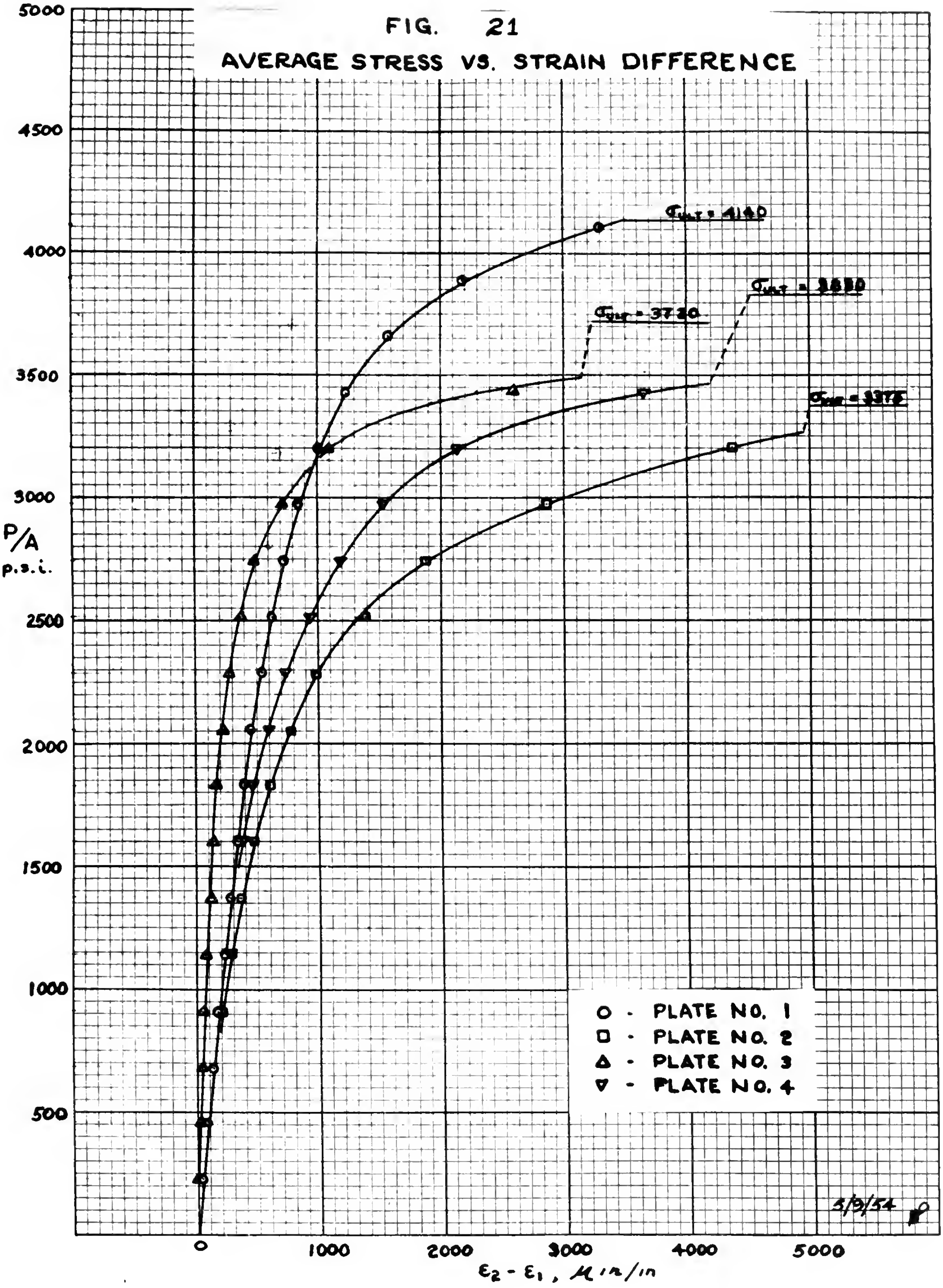


FIG. 21

AVERAGE STRESS VS. STRAIN DIFFERENCE

IV. DISCUSSION OF RESULTS

A. Achievement of Uniform Stress

1. General

Figures 1 to 20 represent the results of an effort to obtain an acceptable stress distribution along the loaded edge of the test plate. Test No. 1 (Fig. 1) indicated immediately that the equipment was not performing as desired. The values of the average strains did however tend to vary about the mean calculated strain indicated by the dashed line in Figs. 1 to 20 inclusive. Subsequent tests are attempts to isolate various causes for the wide variation in strain. It must be admitted that the results of these further tests were not completely satisfactory. Fairly wide strain variation continued to exist in all of the tests. A summary of comparison of average observed to average computed strains and percent deviation for the various tests appears in Table IIIA, Appendix C. It may be said however that in general the average line of the plotted distribution of the observed strains remained parallel to the loaded edge. The necessity of using the hydraulic jacks has not been clearly demonstrated here. Test Nos. 15 & 16 (Figs. No. 17 & 18) show scatter not far different from other tests except that a load concentration appeared on one "a"-edge. Such results however depend on the accuracy to which the I-beams can be shimmed into a parallel position. Under heavier loads that were not needed in these tests but which will be required for completion of the testing program, the local stress concentrations produced by the beam shims may affect the I-beams locally and destroy the original alignment. Such action was actually observed when the I-beams were originally set up and placed in

IV. DISCUSSION OF RESULTS

A. Achievement of Uniform Stress

1. General

Figures 1 to 20 represent the results of an effort to obtain an acceptable stress distribution along the loaded edge of the test plate. Test No. 1 (Fig. 1) indicated immediately that the equipment was not performing as desired. The values of the average strains did however tend to vary about the mean calculated strain indicated by the dashed line in Figs. 1 to 20 inclusive. Subsequent tests are attempts to isolate various causes for the wide variation in strain. It must be admitted that the results of these further tests were not completely satisfactory. Fairly wide strain variation continued to exist in all of the tests. A summary of comparison of average observed to average computed strains and percent deviation for the various tests appears in Table III, Appendix C. It may be said however that in general the average line of the plotted distribution of the observed strains remained parallel to the loaded edge. The necessity of using the hydraulic jacks has not been clearly demonstrated here. Test Nos. 16 & 15 (Figs. No. 17 & 15) show another one far different from other tests except that a load concentration appeared on one "A"-edge. Such results however depend on the accuracy to which the I-beams can be obtained into a parallel position. Under heavier loads that were not needed in these tests but which will be required for completion of the testing program, the local stress concentrations produced by the beam shims may affect the I-beams locally and destroy the original alignment. Such action was actually observed when the I-beams were originally set up and placed in position. Also, the loads used in this part of the project were not severe enough to materially deflect the I-beams. It is this deflection which the use of the jacks was designed to overcome. It was expected that on the plates with shorter "b"-edge dimensions, the use of jacks would not be necessary. Based on the results of this thesis, it is felt that no conclusive statement on the future use of the jacks should be made.

2. Strain Gage Locations

The strain gage locations on Test Plate No. 1 (Fig. 1B) were placed as close to the edge of its plate as practicable as it was expected that at this location the strain variation would be a maximum. Due to the wide variations in strain observed, it is believed that further strain readings are needed from more closely spaced gages between jack positions in order to trace the reasons for these variations. It should be pointed out that where the 3/4" loading bar is used, peak strain readings do not always fall under jack locations. Also the jack foot itself (see Fig. 27) is not solid across the loading face. This may give rise to some strain variations especially when lighter loading bars are used.

It is expected that the strains at the middle of the test plate would be considerably moderated through shear action even when varying adjacent strains are recorded near the loaded edges. It is the strain at or near the center of its plate that is of primary interest when tests or buckling are performed. Further experiments should be performed to determine the validity of this assumption.

3. Performance of Hydraulic Jacks

It is known that two factors are present which could affect the position. Also, the loads used in this part of the project were not severe enough to materially deflect the I-beam. It is this deflection which the use of the jacks was designed to overcome. It was expected that on the plates with shorter "bridge" dimensions, the use of jacks would not be necessary. Based on the results of this study, it is felt that no conclusive statement on the future use of the jacks should be made.

2. Strain Gage Locations

The strain gage locations on Test Plate No. 1 (Fig. 18) were placed as close to the edge of the plate as practicable as it was expected that at this location the strain variation would be a maximum. Due to the wide variations in strain observed, it is believed that further strain readings are needed from more closely spaced gages between jack positions in order to trace the reasons for these variations. It should be pointed out that where the 3/4" loading bar is used, peak strain readings do not always fall under jack locations. Also the jack foot itself (see Fig. 21) is not solid across the loading face. This may give rise to some strain variations especially when lighter loading bars are used.

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3. Performance of Hydraulic Jacks

It is known that two factors are present which could affect the uniformity of action of the hydraulic jacks. Friction forces may vary from jack to jack and probably vary considerably within one jack with changing loads. The plunger return springs in each jack may not produce identical return forces. For normal jack usage, the uniformity of the plunger return springs is of little consequence, hence it cannot be assumed that the manufacturer has taken special care to obtain such uniformity.

The results of Tests 3 to 6 (Figs. 3 to 6) could indicate that the lower jack system was performing better than the upper system since the variations in strain always appeared to be less on the lower plate edge. In order to check this indication and to check the jack performance as a whole, Tests Nos. 10 to 12 (Figs. 10, 12, & 13) were run with a loading bar 1/16" in thickness. This bar merely served to hold the loading segments in place between the jacks. Very little load distribution by this bar was expected or found. Referring to Fig. 13 it would appear that the jacks functioned uniformly. When considering that the possible error in each of the strain points plotted is of the order of $\pm$ 5 microinches per inch, these results are excellent evidence that the jacks are functioning properly. The low order of the readings compared to the computed average strain has no particular significance here since the strain readings under each jack location are also under the hollow part of the jack foot. Maximum readings would occur about one and one-half inches on either side of the present gage location depending precisely on the relative position of each segment in the critical area.

4. Effects of Gaskets

A comparison of Figs. 1, 2, 3, & 16 can be made to study the effect uniformity of action of the hydraulic jacks. Variation forces may vary from jack to jack and probably vary considerably within one jack with changing loads. The plunger return springs in each jack may not produce identical return forces. For normal jack usage, the uniformity of the plunger return springs is of little consequence, hence it cannot be assumed that the manufacturer has taken special care to obtain such uniformity.

The results of Tests 4 to 6 (Figs. 3 to 6) could indicate that the lower jack system was performing better than the upper system since the variations in strain always appeared to be less on the lower plate edge. In order to check this indication and to check the jack performance as a whole, Tests Nos. 10 to 12 (Figs. 10, 11, & 13) were run with a loading bar 1/16" in thickness. This bar merely served to hold the loading segments in place between the jacks. Very little load distribution by this bar was expected or found. Referring to Fig. 13 it would appear that the jacks functioned uniformly. When considering that the possible error in each of the strain points plotted is of the order of ± 5 microinches per inch, these results are excellent evidence that the jacks are functioning properly. The low order of the readings compared to the computed average strain has no particular significance here since the strain readings under each jack location are also under the hollow part of the jack foot. Maximum readings would occur about one and one half inches to either side of the present gage location depending precisely on the relative position of each segment in the vertical beam.

4. Effects of Gaskets

A comparison of Figs. 4, 6, 7, & 14 can be made to study the effect of placing gasket material along the loaded edges of the plate. From Figs. 1 & 2 it can be seen that the use of a neoprene gasket materially reduced the strain variation. Comparing Fig. 3 with Fig. 2 it appears that the use of a single solder wire gasket further reduces these variations to a small extent. Figs. 3 and 16 show little change resulted from using a double rather than a single solder strip. This comparison is complicated by the fact that the flat side of the loading bar was used in Fig. 16.

It seems evident that part of the strain variations are due to local unevenness in the plate or loading bar and that the effect of these points on the strain distribution can be reduced through the use of a gasket material. The solder gasket appears to be slightly more beneficial than a neoprene gasket although this difference may have disappeared entirely if a thicker neoprene gasket were used. The results also tend to indicate the advisability of using new gasket material for each test.

5. Size and Shape of the Loading Bar

A comparison of Figs. 15 and 16 indicated that the presence of the 3/4" thick loading bar quite logically tends to bridge the gap between the points of load application under each jack but in so doing introduces local strain variation of considerable magnitude.

An attempt was made to determine if an optimum bar size existed between the two extreme sizes used in Figs. 15 and 16 by inserting bars of intermediate size. (See Figs. 19 & 20.) The results of these tests seem to indicate that the thicker bar should give better results but that the 3/8" x 1 1/4" bar gives acceptable results. Since the plate tested uses the widest jack spacing of all the plates in the test program, of placing gasket material along the loaded edges of the plate. From Figs. 1 & 6 & 19 can be seen that the use of a neoprene gasket materially reduced the strain variation. Comparing Fig. 3 with Fig. 2 it appears that the use of a single solder wire gasket further reduces these variations to a small extent. Figs. 3 and 16 show little change resulted from using a double rather than a single solder strip. This comparison is complicated by the fact that the flat side of the loading bar was used in Fig. 16.

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When using the 3/4" loading bar a choice existed on which face to use, flat or grooved. During the early tests it was felt that the position of the segments in the groove may have given rise to local strain variations. Hence, the flat side of the bar was used in subsequent tests. A comparison of Figs. 3 & 4 however shows that little difference in strain variation can be attributed to this effect. Actually during the buckling tests, it was found that it is necessary to use the grooved side of the loading bar to prevent lateral movement of the loaded edges when high deflections occur.

6. Effect of Shims

A study of Figs. 4, 5, 9, & 16 seems to indicate that the presence of shims has little effect on the strain variation. However it is felt that shims are necessary when conducting buckling tests. The milled grooves in the loading segments are approximately 1/32" wider than the plate thickness. The presence of the shims decreases the possibility of eccentric loading on the plate by insuring that the plate edge is centered at or under the radial center of the loading segment. It further aids in straightening the plate edge when the grooved side of the loading bar is used.

7. Effect of Loading Segments

Two tests (Figs. 7 & 8) were run without segments but with a sheet lead gasket.between the plate and the loading bar. Strain variations for these tests were generally worse than for other comparable runs. It is

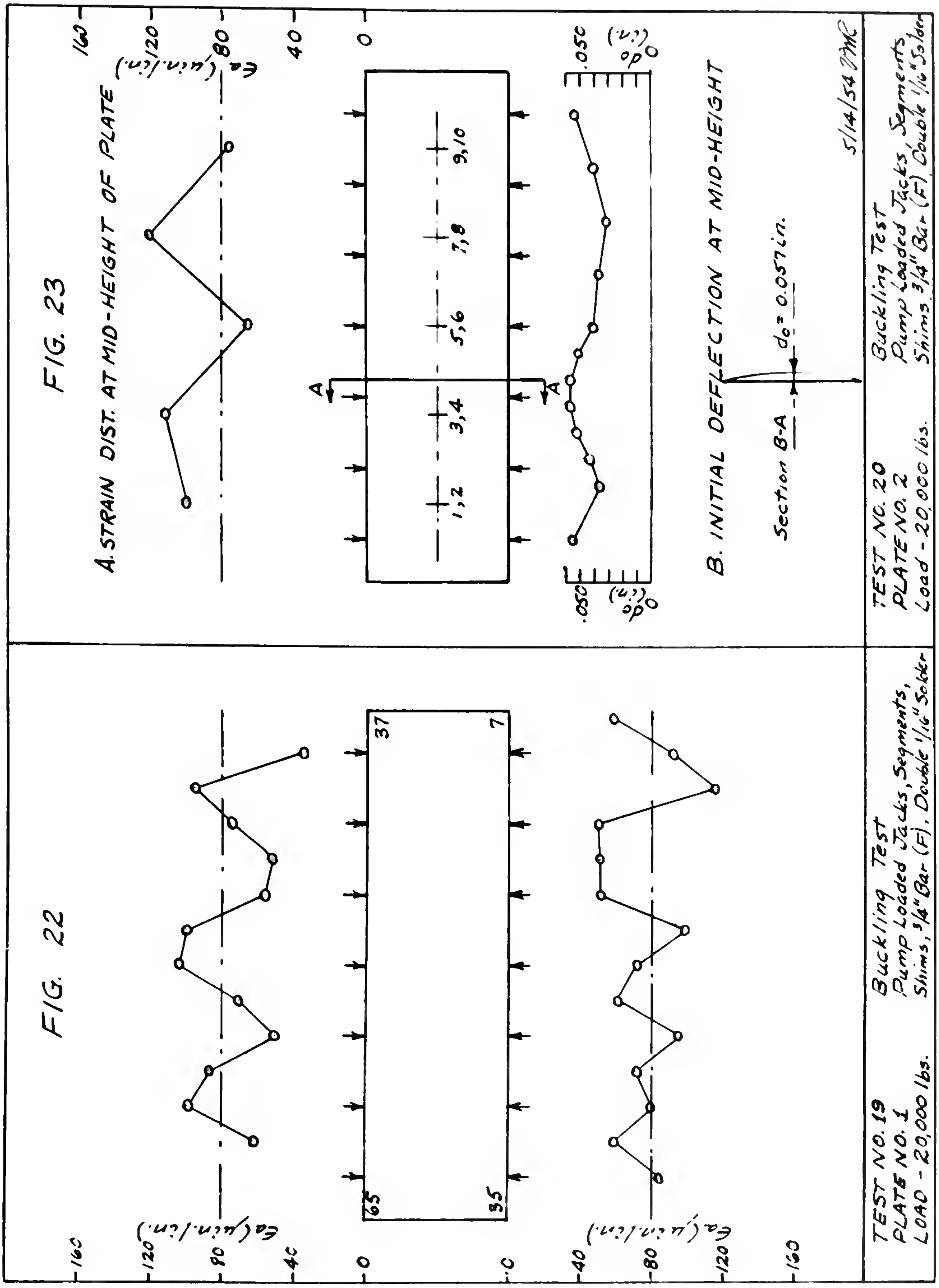

FIG. 22
ϵ_a (μin./in.)
160
120
80
40
0
65
37
35
7
0
40
80
120
160
TEST NO. 19
PLATE NO. 1
LOAD - 20,000 lbs.
Buckling Test
Pump Loaded Jacks, Segments,
Shims, 3/4" Bar (F), Double 1/16" Solder
FIG. 23
A. STRAIN DIST. AT MID-HEIGHT OF PLATE
160
120
80
40
0
A
1,2
3,4
5,6
7,8
9,10
.050
d_o (in.)
B. INITIAL DEFLECTION AT MID-HEIGHT
Section B-A
d_o = 0.057 in.
5/14/54 VMC
TEST NO. 20
PLATE NO. 2
Load - 20,000 lbs.
Buckling Test
Pump Loaded Jacks, Segments
Shims 3/4" Bar (F) Double 1/16" Solder

difficult to separate the effects of the change in gasket material and the effect of the segments. The combination of effects however causes higher strain variations. It was felt that the gasket material used here (lead roof flashing) seems too stiff a material and its effect masked that of the segments.

B. Buckling Data

1. General

The comparison of the critical stress of the plate specimens with that of the theoretical predictions (See Table IIIB) was most gratifying and constituted an important phase in the overall evaluation of the test equipment. Although evidence of a concrete nature is lacking these results tend to support the view that both the stress distribution and the achievement of simple support on the plate edges was satisfactory at least to a practical degree. Further tests should be run to support this view. In particular a separate test perhaps without "a"-edge supports should be run to check the action of the segments to determine if rotational restraint is present. It is felt that fairly high loading would have to be developed before any appreciable resisting moment could develop.

The results of the buckling tests do not completely allay the apprehensions developed under the discussions of uniform edge loading. These results do, however, provoke the thought that the degree of uniformity necessary for uniform buckling results is not as stringent as heretofore supposed.

2. Effect of Initial Deflections

It will be noted in the results (Table II), test Plate No. 1 failed

at a considerably higher load than the remaining specimens. It should also be noted that the original ultimate load before the plate was straightened was 1200 lbs. lower than the final ultimate load. Unfortunately too few strain readings were recorded preceding the first ultimate load on which to base an estimate of the critical load. It is believed however that the difference in the first and final ultimate loads can be explained at least in part by the fact that in restraightening the plate, a complex contour of the plate surface was introduced. It is believed that the strength of such a plate is increased depending on the amount and character of the departure of the initial unfairness from the final deflected shape of the plate prior to yielding. With simple support such shape is usually sinusoidal in form. No particular correlation between initial maximum deflection and final ultimate or buckling loads could be found with the remaining plates. This tends to support the belief that in thin plates character of the initial contour is an overriding effect when considering the strength of such plates.

3. Action of Wide Thin Plates Under Edge Compressions

A comparison of Euler's column formula and the solution of the plate equation for the critical stress indicate that the only difference in form except for the Poisson effect in plates is the value of "k" in the solution for plates. For wide plates this value is very close to one. Hence one would expect wide plates to behave very much like columns up to the critical stress. An inspection of Figs. 24, 25, & 26 indicates that its test plates behaved very much like eccentrically loaded columns where the deflection grows continually under increasing load. Under such conditions it is somewhat an artificiality to attempt to define a critical at a considerably higher load than the remaining specimens. It should also be noted that the original ultimate load before the plate was straightened was 1800 lbs., lower than the final ultimate load. Unfortunately too few strain readings were recorded [illegible] ultimate load on which to base an estimate of the [illegible]. It is believed however that the difference in the first and [illegible] loads can be explained at least in part by the fact that [illegible] ing the plate, a complete [illegible]. It is believed that the strength of such plates is increased depending on the amount and direction of the departures of the initial [illegible] from the final deflected shape of the plate prior to yielding. [illegible] simple support such shape is usually sinusoidal [illegible] correlation between initial maximum deflection and [illegible] buckling loads could be formed with the remaining plates. [illegible] support the belief that in thin plates [illegible] is an overriding effect when considering the [illegible]

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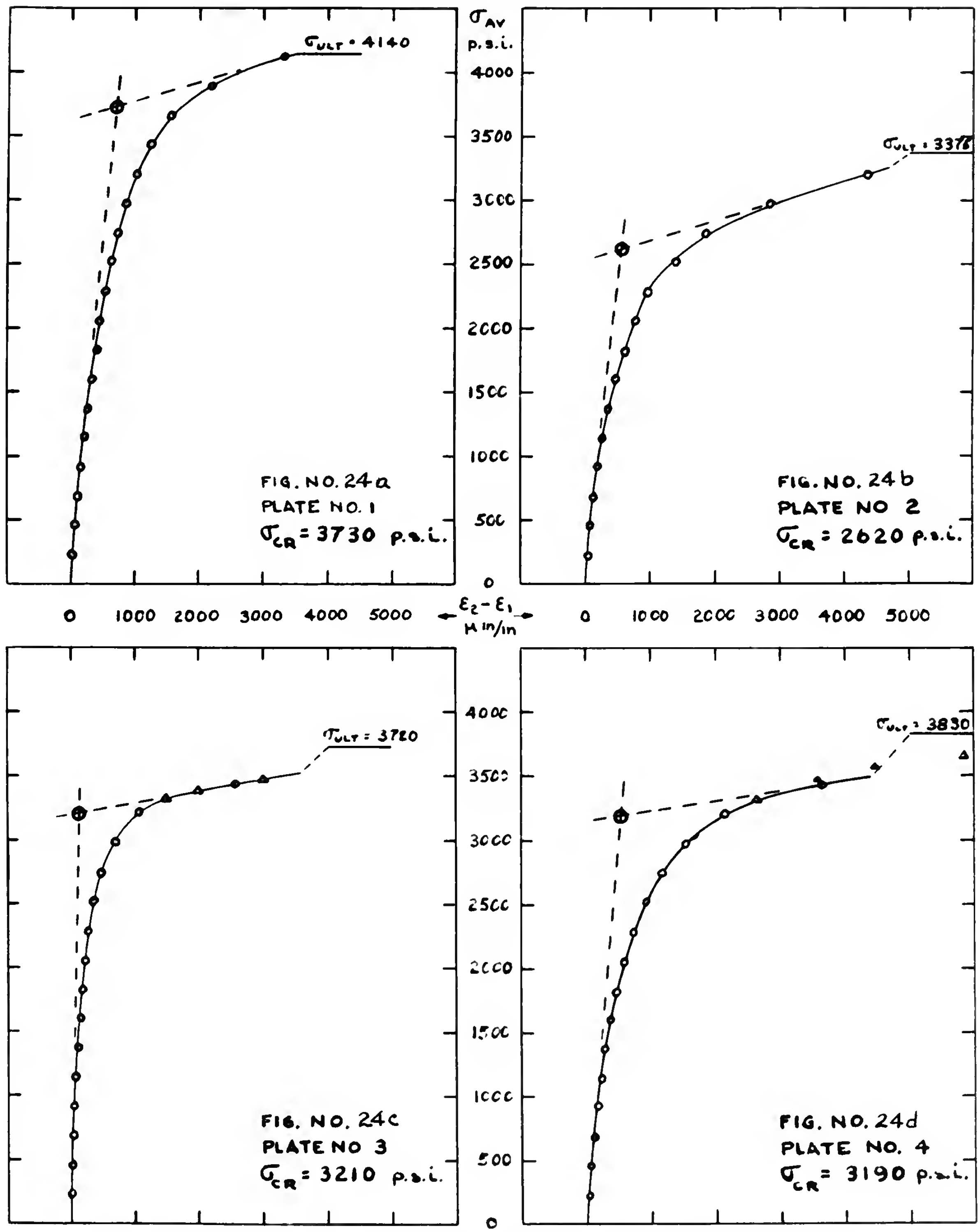
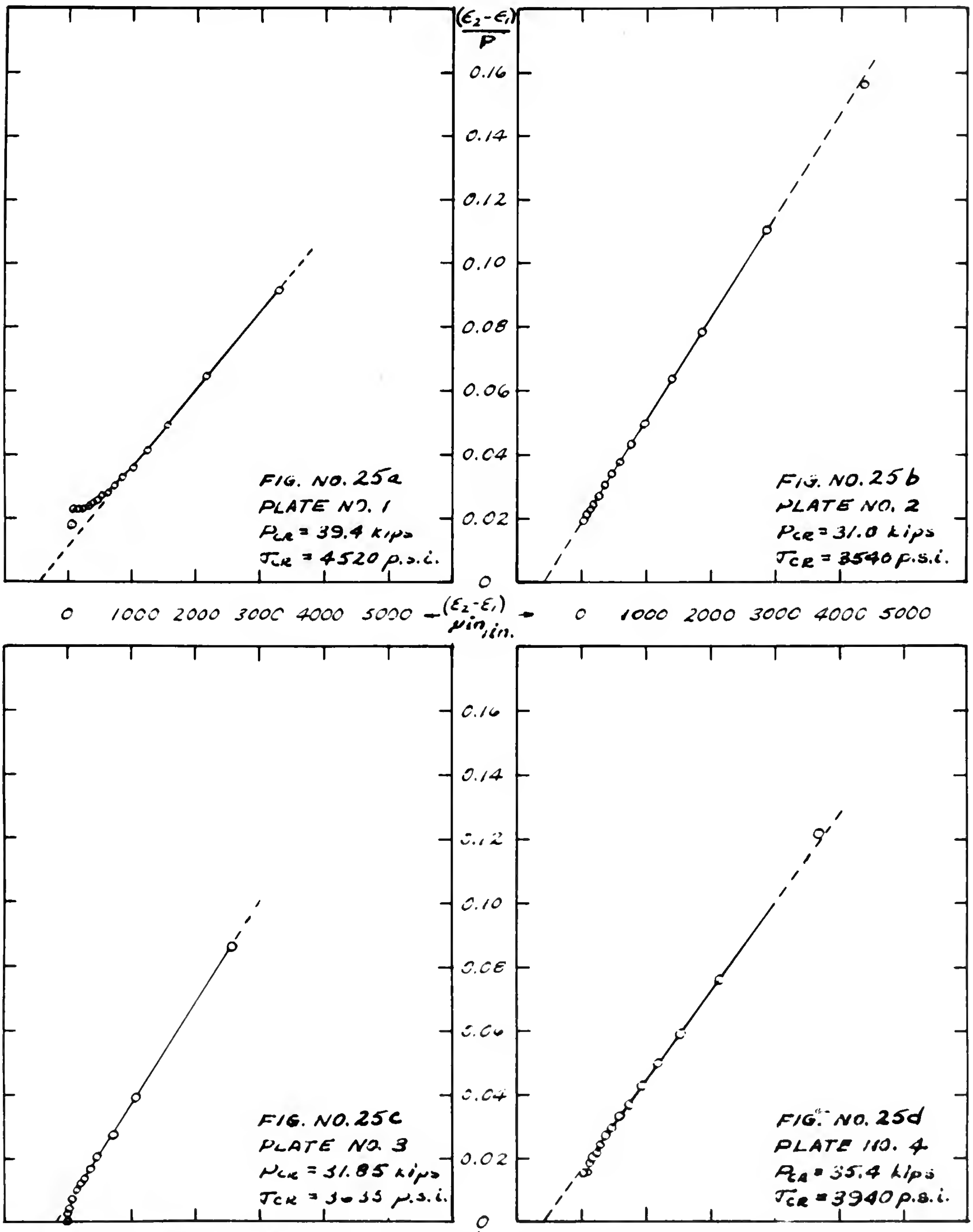
FIG. 24
STRESS VS. STRAIN DIFFERENCE



DETERMINATION OF σ_{cr} BY TOP-OF-THE-KNEE METHOD

5/13/54 M

FIG. 25
STRAIN DIFFERENCE - LOAD RATIO VS. STRAIN DIFFERENCE

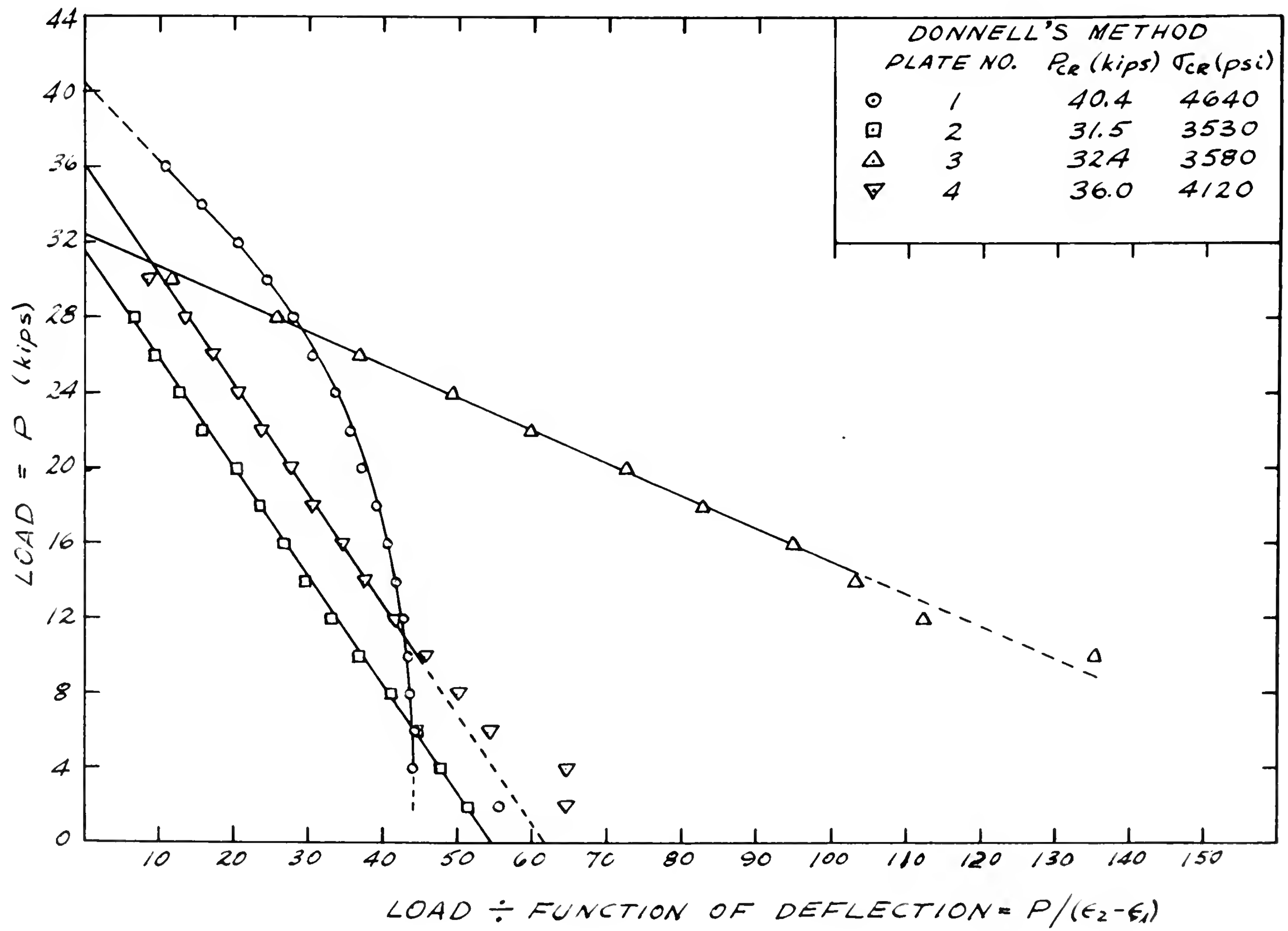


DETERMINATION OF σ_{CR} BY SOUTHWELL'S METHOD

5/14/54

FIG. 26

LOAD VS. LOAD-DEFLECTION RATIO



buckling load. However the necessity exists if comparison to the theoretical buckling stress is to be made. The significant load however, in this case is the ultimate load, the occurence of which is probably coincidental with the development of a plastic hinge in an element of the plate. Bleich [Ref. 8, p. 475] gives a predicted value for the ultimate stress in wide plates based on Marguerre's large deflection theory. Such predicted values based on σ_y and σ_{cr} lead to absurd values for the plates tested since the assumption that the plate remains a rectangle is not valid for these plates. It is believed however that Bleich's predicted values of the critical stress for the remaining plates in the test program will be extremely valuable in alerting the experimentor to impending failure. It should be noted that the authors failed to obtain a sufficient number of readings just prior to the ultimate load on the plate. The character of the load-deflection near failure plays a large part in determining the theoretical value of the critical stress.

4. Calibration of the Testing Machine (See Appendix E).

While this topic is not strictly in the nature of a result it does have a bearing on the validity of the comparison of buckling results. The difference between the load indicated by the testing machine and that indicated by the hydraulic jack pressure at the manifold does not necessarily prove an error exists but it does admit the possibility of such an error. It is expected however that any correction if required will be small since the three gages on the machine agree very closely. Before future testing proceeds however, the testing machine should be calibrated.

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V. CONCLUSIONS

1. The load distribution provided by the test apparatus is not completely satisfactory; however certain lesser conclusions can be drawn.

   a. The hydraulic jacks functioned as they were expected to, although there is not enough data at present to conclude they are necessary.

   b. There was no general slope to the mean line of strain distribution; rather the strains varied about the computed average value.

   c. The stiffness of the loading bar is an important factor in the load distribution and one which is subject to more study.

   d. Factors other than the jacks produced serious strain variations as is evidenced by the fact that the strain peaks showed no consistent tendency to occur under the jacks when the heavier bars were used.

   e. Load distribution is not markedly influenced by type of gasket and whether or not shims are used; however a new gasket should be used for each test and shims should be used to reduce eccentricity of loading.

2. Although no specific investigation was made, it is tentatively concluded that simple support was obtained, in view of the following observations:

   a. The final buckled configuration.

   b. The fact that the edges remained straight even after buckling, especially when the grooved bar was used.

   c. The resultant critical stresses.

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2. Although no specific investigation was made, it is tentatively concluded that simple support was obtained, in view of the following observations:

   a. The flask bowed uniformly.

   b. The fact that the edges remained straight even after cracking, especially when no [illegible] was used.

   c. The resultant cracking [illegible].

3. The equipment is essentially satisfactory for proceeding with future testing.

4. The grooved bar should be used for buckling tests.

5. The stress distribution at the edge of the plate is not extremely critical for buckling tests.

6. The buckling and ultimate strength of wide thin plates is not a function of the maximum measured unfairness alone since the initial contour and past history of the plate may have overriding effects.

7. Due to the difficulty of keeping initial deflections small with respect to the thickness for these plates, buckling loads cannot be obtained directly but must be obtained by extrapolating the data.

8. Wide thin plates act like columns and their strength is affected but little by the presence of support at the unloaded edges.

9. Contingent upon the previous conclusions, valuable buckling data was obtained for one series of plates.

10. The theory as presented in Bleich [8] is supported, and can be used at least to predict the approximate critical loads for future experimental works.

11. Insufficient data was taken in the vicinity of the critical load.

12. Insufficient data was obtained to analyze the growth of unfairness in the plates.

13. While no conclusive evidence exists, there is reasonable doubt as to the accuracy of calibration of the testing machine.

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13. While no conclusive evidence exists, there is reasonable doubt as to the accuracy of calibration of the testing machine.

VI. RECOMMENDATIONS

1. Further investigation of the stress distribution should be made, and attempts made to improve it if necessary, using a plate with a larger thickness and smaller a/t ratio. Strains should be measured at one edge (as before), at the center line, and in detail between two adjacent jacks both at the edge and the centerline to determine more completely the nature of the distribution.
2. Further tests should be conducted to compare the operation of the apparatus when the hydraulic jacks are used and when the loading bar bears directly upon the I-beam.
3. Column tests should be run (no "a"-edge supports) to determine whether simple support is obtained at the "b" edge.
4. The test plates will be more accurately cut if sandblasting is done before the edges are machined instead of after.
5. Bleich's theory should be utilized to predict the critical load for future experimental work.
6. More data should be obtained in the region of the critical load.
7. A dial gage with a longer travel is necessary to measure the growth of unfairness as the plate is loaded.
8. Before more tests are made, the testing machine should be calibrated.

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2. Further tests should be conducted to compare the operation of the apparatus when the hydraulic jacks are used and when the loading bar bears directly upon the I-beam.
3. Column tests should be run (as "s"-edge supports) to determine whether simple support is obtained at the "s" edge.
4. The test plates will be more accurately cut if machining is done before the edges are machined instead of after.
5. Bleich's theory should be utilized to predict the critical load for future experimental work.
6. More data should be obtained in the region of the critical load.
7. A dial gage with a larger travel is necessary to measure the growth of waviness as the plate is loaded.
8. Before more tests are made, the testing machine should be calibrated.

VII. APPENDIX

A. SUPPLEMENTARY INTRODUCTION

1. S-9 Project

Presented herein is a pertinent excerpt of Project S-9 of the Hull Structures Committee of the Society of Naval Architects and Marine Engineers.

Proposal

Project S-9, Buckling Strength of Hull Structures

9 April 1953

"It is the opinion of the S-9 panel that progress in the subject of the buckling strength of hull structures may best be accomplished by a combination of library and experimental research preferably carried out together to avoid duplication and, at the same time, to avoid undue delay. The two proposals presented at the 5 December meeting of the Hull Structure Committee possessed this balance. Since the funds made available do not permit carrying out the dual program proposed, a more modest one based on the same considerations is submitted at this time.

The desired area of study concerns the behavior of elemental flat plates under pure compressive loading. Even for such geometrically simple elements and loadings there is almost no known experimental verification of theory in the range characteristic of transversely framed ship proportions. Witness to this fact is borne out by the extensive use made of Montgomerie's formulations presented before the Society of Naval Architects of Japan in 1934, which nevertheless do not approximate ship conditions in that the short unloaded edges of the plates were unsupported. Furthermore, in this range any data distinguishing between buckling and ultimate loads is almost non-existent.

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The concrete proposal here made is to obtain experimental data for flat plates simply supported on all four edges and subsequently for loaded edges elastically supported with unloaded edges simply supported for side ratios (a/b) of .33, .50, .75 and width-thickness ratios (b/t) of 40, 70 and 100. Elastic edge restraint coefficients of .25 and .50 per page 436 of Bleich's 'Buckling Strength of Metal Structures,' are contemplated for elastic cases.

Following a thorough literature search of this restricted area of the buckling field, such tests might readily be performed on plates 1/8" thick by 3 ft. wide (about 1/4 full size)."

/s/ J. Harvey Evans
Chairman
Panel for Project S-9

2. Buckling Theory

The fundamental general equation for a thin flat plate under the action of forces in its middle plane, when the deflection w is small compared to the thickness t, has been developed by St. Venant from a strain energy analysis approach.

$$\frac{EI}{1-\nu^2}\left(\frac{\partial^4 w}{\partial x^4} + 2\frac{\partial^4 w}{\partial x^2 \partial y^2} + \frac{\partial^4 w}{\partial y^4}\right) + t\left(\sigma_x \frac{\partial^2 w}{\partial x^2} + \sigma_y \frac{\partial^2 w}{\partial y^2} + 2\tau_{xy}\frac{\partial^2 w}{\partial x \partial y}\right) = 0 \qquad (1)$$

where σ_x and σ_y are the normal stresses in the x and y directions and τ_{xy} is the shear stress. The moment of inertia of a unit strip of plating is $I = \frac{t^3}{12}$, E is Young's modulus, and ν is Poisson's ratio.

When $\sigma_y, \tau_{xy} = 0$, the equation simplifies to the form,

$$\frac{EI}{1-\nu^2}\left(\frac{\partial^4 w}{\partial x^4} + 2\frac{\partial^4 w}{\partial x^2 \partial y^2} + \frac{\partial^4 w}{\partial y^4}\right) + \sigma_x t \frac{\partial^2 w}{\partial x^2} = 0 \qquad (2)$$

Bleich [8] has suggested an empirical modification of equation (2) for all cases, even those where the critical stress may exclude the proportional limit.

$$D\left(\tau \frac{\partial^4 w}{\partial x^4} + 2\sqrt{\tau}\frac{\partial^4 w}{\partial x^2 \partial y^2} + \frac{\partial^4 w}{\partial y^4}\right) + \sigma_x t \frac{\partial^2 w}{\partial x^2} = 0 \qquad (3)$$

2. Buckling Theory

The fundamental general equation for a thin flat plate under the action of forces in its middle plane, when the deflection w is small compared to the thickness t, has been developed by St. Venant from a strain energy analysis approach.

$$\frac{EI}{1-\nu^2}\left(\frac{\partial^4 w}{\partial x^4} + 2\frac{\partial^4 w}{\partial x^2 \partial y^2} + \frac{\partial^4 w}{\partial y^4}\right) + t\left(\sigma_x \frac{\partial^2 w}{\partial x^2} + \sigma_y \frac{\partial^2 w}{\partial y^2} + 2\tau_{xy}\frac{\partial^2 w}{\partial x \partial y}\right) = 0 \qquad (1)$$

where σ_x and σ_y are the normal stresses in the x and y directions and τ_{xy} is the shear stress. The moment of inertia of a unit strip of plating is $I = \frac{t^3}{12}$, E is Young's modulus, and ν is Poisson's ratio.

When σ_y, $\tau_{xy} = 0$, the equation simplifies to the form:

$$\frac{EI}{1-\nu^2}\left(\frac{\partial^4 w}{\partial x^4} + 2\frac{\partial^4 w}{\partial x^2 \partial y^2} + \frac{\partial^4 w}{\partial y^4}\right) + \sigma_x t \frac{\partial^2 w}{\partial x^2} = 0 \qquad (2)$$

Bleich [3] has suggested an empirical modification to equation (2) for all cases, even those where the critical stress may exceed the proportional limit.

$$D\left(\tau \frac{\partial^4 w}{\partial x^4} + 2\sqrt{\tau}\frac{\partial^4 w}{\partial x^2 \partial y^2} + \frac{\partial^4 w}{\partial y^4}\right) + \sigma_x t \frac{\partial^2 w}{\partial x^2} = 0 \qquad (3)$$

For wide plates, however, he recommends the use of a simplified equation by Ros and Eichinger,

$$D\tau\left(\frac{\partial^4 w}{\partial x^4}+2\frac{\partial^4 w}{\partial x^2\partial y^2}+\frac{\partial^4 w}{\partial y^4}\right)+\sigma_x t\,\frac{\partial^2 w}{\partial x^2}=0 \qquad (4)$$

since it presents an easier solution, and since the difference between the results of equations (3) and (4) is not very great for such plates.

The general solution of equation (4) is

$$w=\sin\frac{\pi y}{b}\left(c_1\cos\mathcal{X}_1\frac{x}{b}+c_2\cos\mathcal{X}_2\frac{x}{b}\right) \qquad (5)$$

where $\mathcal{X}_{1,2}=\frac{\pi}{2}\left(\sqrt{K}\pm\sqrt{K-4}\right)$

and it develops that when the boundary conditions of simple support are satisfied, the following characteristic equation is obtained:

$$\mathcal{X}_1\tan\frac{\alpha\mathcal{X}_1}{2}-\mathcal{X}_2\tan\frac{\alpha\mathcal{X}_2}{2}+\alpha\rho\left(\mathcal{X}_1^2-\mathcal{X}_2^2\right)=0 \qquad (6)$$

where ρ is the coefficient of edge restraint.

This equation yields characteristic values of σ_x at which the plate changes from one configuration to another, the lowest value being the buckling stress, σ_{cr}.

$$\sigma_{cr}=\frac{\pi^2E\tau}{12(1-\nu^2)}\left(\frac{t}{b}\right)^2K \qquad (7)$$

or

$$\sigma_{cr}=\frac{\pi^2E\tau}{12(1-\nu^2)}\left(\frac{t}{a}\right)^2\bar{K} \qquad (8)$$

where $\bar{K}=\alpha^2K.$

For wide plates, however, he recommends the use of a modified equation by Ros and Kleinlogel,

$$D\left(\frac{\partial^4 w}{\partial x^4} + 2\frac{\partial^4 w}{\partial x^2 \partial y^2} + \frac{\partial^4 w}{\partial y^4}\right) + \sigma_x t \frac{\partial^2 w}{\partial x^2} = 0 \qquad (6)$$

since it presents an easier solution, and since the difference between the results of equations (3) and (6) is not very great for such plates.

The general solution of equation (6) is

$$w = \sin\frac{\pi y}{a}\left(C_1 \cos\alpha_1 \frac{x}{b} + C_2 \cos\alpha_2 \frac{x}{b}\right) \qquad (7)$$

where $\alpha_{1,2} = \frac{\pi}{2}\left(\sqrt{k} \pm \sqrt{k-4}\right)$

and it develops that when the boundary conditions of simple supports are satisfied, the following characteristic equation is obtained:

$$\alpha_1 \tan\frac{\alpha_1}{2} - \alpha_2 \tan\frac{\alpha_2}{2} + \zeta\left(\alpha_1^2 - \alpha_2^2\right) = 0 \qquad (8)$$

where ζ is the coefficient of edge restraint.

This equation yields characteristic values of σ [illegible] the plate changes from one configuration to another, the [illegible] the buckling stress, σ_{cr}.

$$\sigma_{cr} = \frac{\pi^2 E}{12(1-\nu^2)}\left(\frac{t}{b}\right)^2 k \qquad (9)$$

$$\sigma_{cr} = \frac{\pi^2 E}{12(1-\nu^2)}\left(\frac{t}{b}\right)^2 \bar{k} \qquad \text{or} \qquad (10)$$

where $\bar{k} = c^2 k$

Bleich presents a table of $\bar{K}$ values for various combinations of and $\int$ [Ref. 8, p. 436]. For plates simply supported .

Solution of equation (3) for conditions of simple support at all four edges gives the result

$$\sigma_{cr} = \frac{\pi^2 E \sqrt{\tau}}{12(1-\nu^2)} \left(\frac{t}{b}\right)^2 \left(\frac{\alpha}{n \sqrt[4]{\tau}} + \frac{n \sqrt[4]{\tau}}{\alpha}\right)^2 \tag{9}$$

where n is the number of half-waves in the buckled shape.

3. Determination of Buckling Load

The theoretical buckling load is the load at which the plate suddenly departs from its plane condition and assumes a buckled shape with two-dimensional curvature. This load can be calculated by applying the "least work" approach introduced by Kirchhoff and applied to plates by Bryan. In practice this phenomenon may not occur abruptly due either to initial "unplaneness" of the plate, or to the fact that the yield point is reached in the fibers of the plate before buckling occurs. The buckling load as obtained in practice must be a rather fictitious value found by some less direct means. Methods available are the "top-of-the-knee" method, the "stress-reversal" method, Vasta's method [19], Yoshiki's method 60 , and Southwell's or Donnell's method [48], [18]. The top-of-the-knee method depends upon the rate of change of the lateral deflection with increasing load. This deflection may be measured directly, or the bending moment in the plate as determined by the difference of the strains on opposite sides of the plate may be used as an indicative value. The strain reversal method defines the buckling load as that at which the slope of the strain-load curve for either side

Bleich presents a table of K values for various combinations of and ρ [Ref. 8, p. 436]. For plates simply supported

Solution of equation (3) for conditions of simple support at all four edges gives the result

$$\sigma_{cr} = \frac{\pi^2 E \sqrt{T}}{12(1-\mu^2)} \left(\frac{t}{b}\right)^2 \left(\frac{\alpha}{n\sqrt[4]{T}} + \frac{n\sqrt[4]{T}}{\alpha}\right)^2 \qquad (9)$$

where n is the number of half-waves in the buckled shape.

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Southwell's method is based upon the bending of a column which has some initial unfairness. Starting with the basic differential equation, the following expression can be arrived at:

$$\frac{d}{P} P_{cr} - d \quad d_o \tag{10}$$

If a plot is made of $\frac{d}{P}$ vs. d , the slope of the line is the inverse of the critical load, and the intercept on the d-axis is the initial deflection d_o.

Donnell has devised a variation on Southwell's method which has some advantages. If P is plotted against $\frac{P}{d}$, a line is obtained whose equation is:

$$P_{cr} \quad P \quad d_o \frac{P}{d} \tag{11}$$

The intercept on the P-axis is the buckling load, and the slope is the initial unfairness.

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$$\frac{d}{P} P_{cr} - d = d_0 \qquad (10)$$

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The intercept on the P-axis is the buckling load, and the slope is the initial unfairness.

B. DETAILS OF PROCEDURE

1. Design of the Test Apparatus

a) General

The survey of the pertinent literature indicated that Polyclrone [40] and Capozzoli [12] had tested plates of high aspect ratios at Massachusetts Institute of Technology. Equipment was designed and built to provide simple support of the unloaded edges. This equipment was made available to this project if its use proved feasible. The total length of support for each edge was $3\frac{1}{4}$"; hence this limited the total length of "a"-edges of each group of plates to be tested if these supports were to be used. A chart was made of the ratios a/b, a/t and "t" of the plate specimens using values of "t" for standard plate sizes and limiting the thickness to 5/16", the maximum capacity of the supports.

The maximum capacity of the materials laboratory hydraulic testing machines was 300,000 pounds. This placed an upper limit on the size of plates which could be tested.

With these two criteria, i.e., the total length of "a"-edges and the testing machine capacity, the plate sizes were selected and are presented in Table I.

b) B-Edges

Since the longest selected plate was 56.75" in length, the testing jig must be about 5 feet in length. The upper head of the testing machine presented an annular bearing surface 18" O.D. by 11" I.D. The lower head was a 2 foot square. Expected load analyses of the proposed plate sizes indicated that size 43.75" x 10.19" x 7/32"

[illegible]

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a) General

The survey of the pertinent literature [illegible] that Polyclrome [40] and Caponi [18] had tested plates of high aspect ratios at Massachusetts Institute of Technology. [illegible] was designed and built to provide simple support of the [illegible] edges. This equipment was made available to the project [illegible] proved feasible. The total length of support [illegible] hence this limited the total length of [illegible] of plates to be tested [illegible] made of the ratios a/b, a [illegible] b of the plate specimens using values of [illegible] for standard plate sizes and [illegible] the maximum capacity of the supports.

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b) S-Edges

Since the [illegible] ing jig must be [illegible] ing machine [illegible]. The lower head was [illegible] proposed plate sizes [illegible] imposed the severest condition on the jig both strength-wise and from the standpoint of distributing the edge loading.

Preliminary calculations indicated that allowing a stress variation of 5% along the loaded edge an I-beam of approximately 12 feet in depth would be required to bridge the gap from the head of the testing machine to 43.75" length of plate edge. This was in line with the experience of the David Taylor Model Basin where constant displacement heads of the same order of magnitude were found necessary.

It was felt that a more practical solution would involve the use of hydraulic means to distribute the load. Accordingly it was found that commercial 20-ton hydraulic jacks could be fitted in the testing jig, seven on each edge connected to a common manifold, which when backed by an I-beam of approximately 18" depth should provide a solution to the problem. These jacks could be placed along the I-beam (See Fig. 27) with different intervals to suit the particular plate size. It was found that these commercial jacks (See Fig. 31) could normally be expected to withstand a 10% overload. Considering this, seven 20-ton jacks would equal the rated capacity of the testing machine, i.e., 300,000 pounds.

There still remained the problem of converting the load from each jack to a more uniform load on the plate edges and still maintaining this edge as simply supported. Frankland [19] merely butted the square edge of the plate against the flat loading lead relying on friction to prevent lateral motion; Yoshiki [60] used the same arrangement except that he rounded the edges of the plate to minimize the rotational restraint.

imposed the severest condition on the jig both strength-wise and from the standpoint of distributing the edge loading.

Preliminary calculations indicated that allowing a stress variation of 5% along the loaded edge an I-beam of approximately 17 feet in depth would be required to bridge the gap from the head of the testing machine to 43.75" length of plate edge. This was in line with the experience of the David Taylor Model basin where constant displacement loads of the same order of magnitude were found necessary.

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There still remained the problem of converting the load from each jack to a more uniform load on the plate edges and still maintaining this edge as simply supported. Frankland [19] merely rested the square edge of the plate against the flat loading head relying on friction to prevent lateral motion; Yoshiki [60] used the same arrangement except that he rounded the edges of the plate to minimize the rotational restraint.

Since both of these methods necessitated very careful machining of the plate edges, it was felt that the same result could be accomplished by using the milled or slotted cylinders which would fit over the edge of its plate and bear on a loading bar which could also be used to bridge the gap between the jacks. (See Fig. 27) The cylinders, or loading segments should be restricted in length to minimize torsional restraint on the loaded plate edge. Both the segments and the bar could then be hardened to resist deformation under the high loading at the point of contact. The loading bar was ground flat on one surface and flat with a shallow groove in the other surface. It was believed this groove might be necessary to restrict lateral movement of the plate.

c) "A"-Edges

In order to avoid transmittal of load to the "a"-edge supports, these supports should be completely independent and mounted free from the "b"-edge loading arrangement. The "a"-edge support should prevent lateral motion of the edge of the plate while permitting both rotation and translation in the plane of the plate.

It was found that the "a"-edge supports designed by Polychrone [40] and modified by Capozzoli [12] permitted freedom of rotation of the plate edge for about 20° on either side of the plate axis if the ball bearing races were permitted to slide on the channel sides (See Fig. 29). The ball bearing support allows translation in the plane of the plate and can be adjusted to the thickness of the plate by the use of spacers between the race support and the channel, with the final adjustment of the desired pressure being accomplished with the thumb screws. The channels are held in place by an arrangement of

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It was found that the "a"-edge supports developed by Wittrick [40] and modified by Capaccioli [12] permitted freedom of rotation of the plate edge for about 30° on either side of the plate axis if the ball bearing races were permitted to slide on the channel sides (See Fig. 29). The ball bearing support allows translation in the plane of the plate and can be adjusted to the thickness of the plate by the use of spacers between the main support and the channel. With the final adjustment of the desired pressure being supplied with the thumb screws. The channels are held in place by set screws …

rods and turnbuckles connected to angle iron braces rigidly attached to the frame of the testing machine. (See Fig. 33)

The two "a"-edges are connected to each other by an angle on either side of the plate to prevent the support from slipping off the plate edge. These angles both have one edge planed to serve as a base for the sliding dial gage with which actual deflection measurements (d') can be read.

d) Hydraulic Jacks

The I-beams were both skim-machined to provide a flat base for the hydraulic jacks. These jacks are further held in position by two angles on either side which are bolted to the I-beam but hold the jacks only by clamping friction. These angles are slotted to accommodate the hydraulic fittings as the jack positions are changed. The Blackhawk Company, manufacturers of the hydraulic jacks, indicated that reasonably equal pressures could be expected in each jack when several were connected to a common manifold. Some slight variation in delivered force, however, may be expected from varying friction loads in the different jacks and varying back pressures created by the tension in the spring of the piston return mechanism.

Each of the seven jacks along one edge of the plate was connected to a manifold manufactured from 3" diameter cold rolled steel. (See Fig. 30). This manifold also had outlets on which a pump and pressure gage could be mounted. (See Fig. 34).

rods and turnbuckles connected to angle iron braces rigidly attached to the frame of the testing machine. (See Fig. 33)

The two "a"-edges are connected to each other by an angle on either side of the plate to prevent the support from slipping off the plate edge. These angles both have one edge planed to serve as a base for the sliding dial gage with which actual deflection measurements (4') can be read.

d) Hydraulic Jacks

The I-beams were both side-machined to provide a flat base for the hydraulic jacks. These jacks are further held in position by two angles on either side which are bolted to the I-beam but hold the jacks only by clamping friction. These angles are slotted to accommodate the hydraulic fittings as the jack positions are changed. The Blackhawk Company, manufacturers of the hydraulic jacks, indicated that reasonably equal pressures could be expected in each jack when several were connected to a common manifold. Some slight variation in delivered force, however, may be expected from varying friction loads in the different jacks and varying back pressures created by the tension in the spring of the piston return mechanism.

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2. Procedure for Setting Up Equipment

In order to facilitate future use of the equipment a brief description of the method of assembling the test equipment is given. The weight handling equipment consists of a chain hoist and a narrow four wheeled truck which will fit easily between the frame of the testing machine. The steps are as follows:

a.) With the chain hoist and sling place the upper I-beam on the truck and move into position in the testing machine under the upper head of the machine. A 1/4" shim will be found between a spacing ring clamped to the I-beam and the beam itself. This shim should be towards the window. It is convenient to lift the wheels of the truck over the ledge created by the difference in height of the floor level and the lower head, by means of a lever. The wheels are then blocked up to the proper height and the truck moved on to the lower head. Once the equipment is on the lower head it is advisable to raise the lower head slightly, about 1/2" to prevent the possibility of locking equipment between the upper and lower heads. Should such a lock become positive it might be necessary to dismantle the testing machine to remove the equipment.

b.) The upper head is lowered carefully over the 2" stud protruding from the I-beam. The plate washer and nut is secured loosely on the stud. The I-beam may now be lifted clear of the truck and the truck removed. It may be convenient now to lower its I-beam until it barely rests on the lower head. The beam may now be positioned on the head. Scribe marks will be found on the spacing ring and the upper head of the testing machine to facilitate alignment. When the beam is in place the 2" nut should be tightened and the upper head raised about 2 ft. clear of the lower head.

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c.) The upper system of jacks, already positioned as desired between the clamping angles may now be lifted as a unit and placed on the truck. It will be found that if care is taken, the jack feet are wide enough to prevent this assembly from tipping. The truck is then moved into position under the upper I-beam and the bolts secured. The upper head and jack system is lifted to a maximum height and the truck removed.

d.) It is usually convenient to leave the lower system of jacks attached to the lower I-beam. This whole assembly then may be lifted on the truck and moved into position on the lower head with only the front wheels on the lower head. The lower head is raised about 1" or 2" and a block is placed under the higher end of the beam. The head is then lowered so that one end of the I-beam now rests on the block. The other end of the beam is then lifted clear of the truck with the chain hoist and the truck removed. The lower head is then raised until the I-beam is nearly supported by it. A 1/4" shim is inserted under the edge of the I-beam nearest the window. It is advisable at this point to position the lower I-beam on the head by lowering the upper head and matching the opposing jacks. Such shims as deemed necessary under the lower head are best inserted at this point. The blocks and chain hoist are then removed and the lower head brought down to about 1" from its lowest position.

e.) The "a"-edge supporting angles with rods attached may then be placed and clamped to the proper height on the H-beam frames of the testing machine. The "a"-edge channels, held together with their supporting angles which are spaced according to the width of the plate specimen, are slid into position and rested on the ends of the lower loading bar. The turnbuckles may then be secured and the "a"-edge supports lined up.

c.) The upper system of jacks, already positioned as desired between the clamping angles may now be lifted as a unit and placed on the truck. It will be found that if care is taken, the jack feet are wide enough to prevent this assembly from tipping. The truck is then moved into position under the upper I-beam and the bolts secured. The upper head and jack system is lifted to a maximum height and the truck removed.

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e.) The "a"-edge supporting angles with rods attached may then be placed and clamped to the proper height on the H-beam frames of the testing machine. The "a"-edge channels, held together with their supporting angles which are spaced according to the width of the above specimen, are slid into position and rested on the ends of the lower bearing. The turnbuckles may then be secured and the "a"-edge support lifted up.

f.) After initial deflection of the plate readings are taken, the test plate is placed in position between its "a"-edge channels and rested on the lower loading bar. The ball-bearing races are inserted to hold the plate in place. The shims for the loading segments, if used, are held in position along the top edge either with C-clamps or a segment placed at each end, or both. The gasket material may be placed along the edge and the remaining segments positioned on the top edge of the plate over the gasket and shims. The method of placing the segments along the bottom edge of the plate depends upon the type of gasket material used. The thumbscrews on the "a"-edge channels are tightened so that the plate may be suspended from the "a"-edge supports. The lower head of the testing machine is then lowered slightly. If solder gasket material is used, the head is lowered about 1/2". The segments may then be slid into position from either end of the plate. The solder may also be run in from the end of the plate between the segments and the plate edge. The shims are then inserted between the segment and the side of the plate. If a flat gasket such as the neoprene strip is used the segments must be rolled in from the side of the plate. Careful spacing of the distance between the plate edge and the loading bar is necessary to accomplish this successfully. The lower head is then raised until the plate is supported. Both upper and lower head and the "a"-edge supports are then positioned so that approximately equal plate overhang remains above and below each "a"-edge support. The plate is now ready to load.

g.) The strain gage hook up may now be completed and zero readings taken. The pressure gage and hand pump may be connected to the manifolds if desired. A slight load should be placed on the plate, probably not exceeding 1000 pounds. The loaded edges of the plate may be forced into

a straight line as the segments tend to seek the lowest part of the groove in both loading bars. It is useful to apply some grease to the loading bar grooves to facilitate this operation and subsequent sliding when plate deflections are large. The straightness of the edge may be checked by eye or with a straight edge. Zero or near-zero deflection readings may be taken with the dial gage at desired locations along the horizontal center line of the plate.

h.) The application of the load should be gradual at first until all jacks have time to equalize their internal pressure. There should be about 2" of each jack ram protruding, more perhaps when higher loads are reached. The amount will depend on the degree of leakage in the system. A stress of about 1000 psi is needed to deform the solder gasket if this type is used. The load corresponding to this stress should be placed on the plate and allowed to remain until the rate of decrease of its load is nearly zero. If it is desired to obtain test points below this loading care must be taken not to unload the plate entirely if straightening of the "b"-edge was required. The test is now ready to proceed.

i.) Disassembly of the equipment generally follows the reverse of the method described above.

3. Explanation of Test Description

In order to evaluate the effect of the various components of the test apparatus, tests under different conditions were run and described cryptically in the data sheets and on the graphic results. (Figs. 1 to 20).

a straight line as the segments tend to seat the lowest part of the groove in both loading bars. It is useful to apply some grease to the loading bar grooves to facilitate this operation and subsequent sliding when plate deflections are large. The straightness of the edge may be checked by eye or with a straight edge. Zero or near-zero deflection readings may be taken with the dial gage at desired locations along the horizontal center line of the plate.

e.) The application of the load should be gradual at first until all jacks have time to equalize their internal pressures. There should be about 2" of each jack ram protruding, more perhaps when higher loads are reached. The amount will depend on the degree of leakage in the system. A stress of about 1000 psi is needed to deform the Teflon gasket if this type is used. The load corresponding to this stress should be placed on the plate and allowed to remain until the rate of decrease of the load is nearly zero. If it is desired to obtain test points below this loading care must be taken not to unload the plate entirely if straightening of the "S"-edge was required. The test is now ready to proceed.

f.) Disassembly of the equipment generally follows the reverse of the method described above.

3. Explanation of Test Description

In order to evaluate the effect of the various components of the test apparatus, tests under different conditions were run and coded cryptically in the data sheets and on the graphic results. (Figs. 1 to 20).

a.) Load Application - Machine or Pump

It was intended that each set of hydraulic jacks (top and bottom) would be connected to a common manifold with the jacks partially extended. The testing machine would then be used to apply the load to the jack system. In an attempt to evaluate effects of jack friction a hydraulic hand pump was connected to the top group of jacks and the load was applied in this manner. It was found also that the quieter operation of the hand pump was more conducive to efficient test procedure, hence this method was generally used on subsequent tests.

b.) Jacks - No Jacks

One set of tests was run without the hydraulic jacks in order to ascertain their effect. When the jacks were removed the two loading I-beams were shimmed so as to be as nearly parallel as practicable.

c.) Bar

The hardened loading bar designed for the apparatus is designated by the thickness (3/4") and whether the flat (F) or grooved (G) side of the bar bore on the segments. (See detail Fig. 27). Since the loading pressures on this particular size of plate are comparatively low, cold rolled steel bars 1 1/4" wide and thicknesses 1/16", 1/4", 5/16" and 3/8" were substituted in various tests in an effort to shed light on the effect of the stiffness of the bar on the test results.

d.) Gasket

In an attempt to eliminate local unevenesses in the edge of the test plate various sizes and types of gasket materials were placed along the edge of the plate under the bearing segments. Where a plastic gasket material was used such as the solder, the gasket was first set by running up the load to approximately 20,000 pounds and allowing the gasket material a.) Load Application - Machine or Pump

It was intended that each set of hydraulic jacks (top and bottom) would be connected to a common manifold with the jacks partially extended. The testing machine would then be used to apply the load to the jack system. In an attempt to evaluate effects of jack friction a hydraulic hand pump was connected to the top group of jacks and the load was applied in this manner. It was found also that the quicker operation of the hand pump was more conducive to efficient test procedure, hence this method was generally used on subsequent tests.

b.) Jacks - No Jacks

One set of tests was run without the hydraulic jacks in order to ascertain their effect. When the jacks were removed the bar loading I-beams were shimmed so as to be as nearly parallel as practicable.

c.) Bar

The hardened loading bar designed for the apparatus is designated by the thickness (3/4") and whether the flat (F) or grooved (G) side of the bar bore on the segments. (See detail Fig. 27). Since the loading pressures on this particular size of plate are comparatively low, cold rolled steel bars 1 1/4" wide and thicknesses 1/16", 1/8", 1/4" and 3/8" were substituted in various tests in an effort to shed light on the effect of the stiffness of the bar on the test results.

d.) Gasket

In an attempt to eliminate local unevenness in the edge of the test plate various sizes and types of gasket materials were placed along the edge of the plate under the bearing segments. Where a plastic material was used such as the solder, the gasket was first set by running up the load to approximately 10,000 pounds and allowing the gasket material to flow plastically for several minutes until the rate of drop in the load was approximately equal to that of normal leakage in the hydraulic system.

The neoprene gasket was cut from a sheet 1/16" thick to a width equal to the approximate thickness of the test plate. Two diameters of solder gaskets were used, 1/16" and 3/32". The 1/16" solder was used as a double strip side by side, whereas only a single strip of the 3/32" diameter solder was used. During two tests where no segments were used, a strip of lead house sheathing about 1" wide and 1/16" thick was placed between the loading bar and the edge of the plate.

e.) Segments - No Segments

Since most of the test results indicated the presence of fairly high moments in the plate from the beginning of the tests, it was decided advisable to determine whether or not the bearing segments influenced this phenomenon. Accordingly during several tests these segments were removed and the plate was butted up against a sheet lead gasket on the loading bar.

f.) Shims - No Shims

The segments were made with the slot a minimum of 1/32" wider than the plate. In order to reduce the possibility of eccentric loading and to check the effect of this clearance on the observed moments in the test plate shim stock was inserted to reduce this clearance. On the tests where shims are indicated, shim stock of .015" and .002" were placed between both sides of the segment and the plate making a total addition of shim stock .034" on each "b"-edge. The maximum estimated clearance between the plate and the segments after the introduction of these shims is .003".

to flow plastically for several minutes until the rate of drop in the load was approximately equal to that of normal leakage in the hydraulic system.

The neoprene gasket was cut from a sheet 1/16" thick to a width equal to the approximate thickness of the test plate. Two diameters of solder gaskets were used, 1/16" and 3/32". The 1/16" solder was used as a double strip side by side, whereas only a single strip of the 3/32" diameter solder was used. During two tests where no segments were used, a strip of lead house sheathing about 1" wide and 1/16" thick was placed between the loading bar and the edge of the plate.

e.) Segments - No Segments

Since most of the test results indicated the presence of fairly high moments in the plate from the beginning of the tests, it was decided advisable to determine whether or not the bearing segments influenced this phenomenon. Accordingly during several tests these segments were removed and the plate was butted up against a sheet lead gasket on the loading bar.

f.) Shims - No Shims

The segments were made with the slot a minimum of 1/32" wider than the plate. In order to reduce the possibility of eccentric loading and to check the effect of this clearance on the observed moments in the test plate shim stock was inserted to reduce this clearance. In the tests where shims are indicated, shim stock of .015" and .010" were placed between both sides of the segment and the plate making a total thickness of shim stock .025" on each "V"-edge. The maximum indicated clearance between the plate and the segments after the introduction of these shims is .007".

g.) Load

The load was measured on the most sensitive gage of the testing machine that would record the load. Three gages are available of ranges 0-30,000 pounds, 0-150,000 pounds, 0-300,000 pounds. All gages have adjustable zero readings so that the weight of the apparatus in the machine is compensated except for the upper loading bar and segments.

h.) Pi - Internal Hydraulic Pressure of Top System of Jacks

In order to provide a check on the total load recorded in the testing machine and to aid in determining the force system on the loading bar should this be necessary, a pressure gage 0-1000 pounds was mounted directly on the upper manifold of the top hydraulic jack system and the pressures recorded.

i.) Strain Gage Readings

These readings are direct readings of the strain gage indicator. The index number did not change during any given test and to save space this number is recorded only under the zero reading. The zero reading was taken after the gasket had been set but with the blocks used to minimize the curvature of the test plate in the "b"-edge direction, still in place. Time did not permit complete cycling of the gages on each run but final zero readings were taken periodically to check the instrumentation. The dummy gage was mounted on a plate of the same thickness and same material as the test plate. The leads to the strain gage indicator were about the same length as those from the test plate. The dummy gage was located as close as practicable to the test plate.

j.) Deflections

Initial deflections in the "a"-edge direction were recorded on g.) Load

The load was measured on the most sensitive gage of the testing machine that would record the load. Three gages are available of ranges 0-30,000 pounds, 0-150,000 pounds, 0-300,000 pounds. All gages have adjustable zero readings so that the weight of the apparatus in the machine is compensated except for the upper loading bar end segments.

h.) P1 - Internal Hydraulic Pressure of Top System of Jacks

In order to provide a check on the total load recorded in the testing machine and to aid in determining the force system on the loading bar should this be necessary, a pressure gage 0-1000 pounds was mounted directly on the upper manifold of the top hydraulic jack system and the pressures recorded.

i.) Strain Gage Readings

These readings are direct readings of the strain gage indicator. The index number did not change during any given test and to save space this number is recorded only under the zero reading. The zero reading was taken after the gasket had been set but with the blocks used to minimize the curvature of the test plate in the "x"-edge direction, still in place. This did not permit complete cycling of the gages on each run but final zero readings were taken periodically to check the instrumentation. The dummy gage was mounted on a piece of the same thickness and same material as the test plate. The leads to the strain gage indicator were about the same length as those from the test plate. The dummy gage was located as near as practicable to the test plate.

j.) Deflections

Initial deflections in the "y"-edge direction were measured on

Plates 2, 3, and 4 at 6" intervals from the center line and at the center. Deflections at the center line in the "b"-edge direction were recorded at each gage location on Plates No. 1 and No. 2 except near the "a"-edges on Plate No. 1 where wiring interfered. Here the deflection noted was between the two end gages as noted. Deflections on Plate Nos. 3 and 4 were recorded at the same location as the initial readings. It will be noted that in certain cases the deflections exceeded the maximum travel of one-half inch on the dial gage.

k.) Plate Orientation

In Figs. 1 to 20 the plate orientation is indicated by the number of the strain gage in each corner. This plate was reversed during later tests along a diagonal as a check on effect of unevenesses in the edge of the plate.

l.) Computed Strain

The computed average strain based on a load of 10,000 pounds and the area of the test plate (b' x t) assuming a value of E 28,300 kips/inch^2 is shown as a dashed horizontal line

$$\sigma_{av} = \frac{10\,000}{56.75 \times .15625 \times 28.3 \times 10^6} = 40.3\ \mu\ in/in$$

Since the accuracy of the individual strains plotted is 5 μ in/in only values of the individual strains above 45.3 or below 35.3 should be considered as different from average values.

4. <u>Data on Test Plates</u>

a.) Length - L = 13.3125"

b.) Width - b' = 56.75"

c.) Thickness - This dimension varied slightly among the four test plates. Measurements were taken along the edge with a micrometer. The

Plates 2, 3, and 4 at 6" intervals from the center line and at the center. Deflections at the center line in the "b"-edge direction were recorded at each gage location on Plates No. 1 and No. 2 except near the "a"-edges on Plate No. 1 where wiring interfered. Here the deflection noted was between the two end gages as noted. Deflections on Plate Nos. 3 and 4 were recorded at the same location as the initial readings. It will be noted that in certain cases the deflections exceeded the maximum travel of one-half inch on the dial gage.

k.) Plate Orientation

In Figs. 1 to 20 the plate orientation is indicated by the number of the strain gage in each corner. This plate was reversed during later tests along a diagonal as a check on effect of unevenness in the edge of the plate.

l.) Computed Strain

The computed average strain based on a load of 10,000 pounds and the area of the test plate (b' x t) assuming a value of E 28,500 kips/inch^2 is shown as a dashed horizontal line

$$\sigma_{av} = \frac{10\,000}{56.75 \times .13625 \times 28.5 \times 10^3} = 40.3\,\mu\,\text{in/in}$$

Since the accuracy of the individual strains plotted is 5 μ in/in only values of the individual strains above 45.3 or below 35.3 should be considered as different from average values.

4. <u>Data on Test Plates</u>

a.) Length - L = 13.3125"

b.) Width - b' = 56.75"

c.) Thickness - This dimension varied slightly among the four test plates. Measurements were taken along the edge with a micrometer. The plates were weighed and an average thickness determined assuming a density of steel 0.283 pounds/inches^3. The surface of the plates were slightly pitted from corrosion and shot blasting. The depth of pits was determined fairly accurately by measuring the "a"-edge where the surface had been ground to the depth of the pits. An average depth of pits as determined by this method is 0.002". The pitting is fairly evenly distributed over the surface and it is estimated that an average thickness is to one half of the average depth of pits. However the effective thickness of its plates is considered to be the thickness to the bottom of the pits since the lines of stress will not follow the contour of the pits to a marked degree. This same procedure is followed in calculating the value of the modulus of elasticity (E), hence the effect will tend to cancel in the analysis of plate strength.

Plate No.	Weight (lbs)	tav. $\frac{\text{Wgt.}}{\rho l b}$ (in)	tav. (micr.) (in)	Pits (in)	tav (corrected) (in)
1	34	.159	.1575	.004	.1535
2	34.2	.160	.1585	.004	.1545
3	33.8	.158	.1585	.004	.1545
4	34.8	.163	.1580	.004	.1540

Nominal thickness of the plate is 0.15625"

Since the scale was accurate to 0.5 pounds and its density of the steel was assumed, it is believed the micrometer readings, even though taken only around the edge of the plate, give a more accurate reading of the plate thickness.

d.) Ratios

"B"-edge Dimension: Defined as the length between "a"-edge supports. 1/4" on each end of the plate was allowed beyond the center line of the plates were weighed and an average thickness determined assuming a density of steel 0.283 pounds/inches³. The surfaces of the plates were slightly pitted from corrosion and shot blasting. The depth of pits was determined fairly accurately by measuring the "B"-edge [illegible] surface had been ground to the depth of the pits. An average depth of pits as determined by this method is 0.005". The [illegible] is fairly evenly distributed over the surface and it is estimated that an average thickness is to one half of the average depth of pits. However, the effective thickness of the plates is considered to [illegible] the bottom of the pits since the lines of stress will not follow the contour of the pits to a marked degree. This [illegible] in calculating the value of the modulus of elasticity, E, [illegible] effect will tend to cancel in the analysis of [illegible].

Plate No.	Weight (lbs)	Ave. $\frac{Wt}{Area}$ (in)	Ave. (Edge) (in)	Pits (in)	Ave (corrected) (in)
1	31	.114	[illegible]	[illegible]	[illegible]
2	34.2	.114	[illegible]	[illegible]	[illegible]
3	[illegible]	.112	[illegible]	[illegible]	[illegible]
4	34.6	.113	.1150	[illegible]	[illegible]

Nominal thickness of the plate is 0.125"

Since the scale was accurate to 0.1 pounds and [illegible] steel was assumed, it is believed [illegible] taken only around the edge of the plate, [illegible] of the plate thickness.

d.) Notation

"B"-edge Dimensions: Defined as the length between [illegible] 1/4" on each end of the plate was allowed [illegible] row ball bearings in the "a"-edge support.

$$b = b' - 2 \times 1/4" = 56.25"$$

"A"-edge Dimension: It was considered originally that this dimension should be the over-all width of the plate plus the distance added by inserting the loading segments over the plate edge. The plate sizes were determined on this basis. Further reflection indicated that the true length of the "a"-edge dimension should not include the addition due to the segments. The edge of the plate extends to the center of the segment (minus gasket thickness). The load however is always applied through this center even after deformation since the segment is circular in cross section.

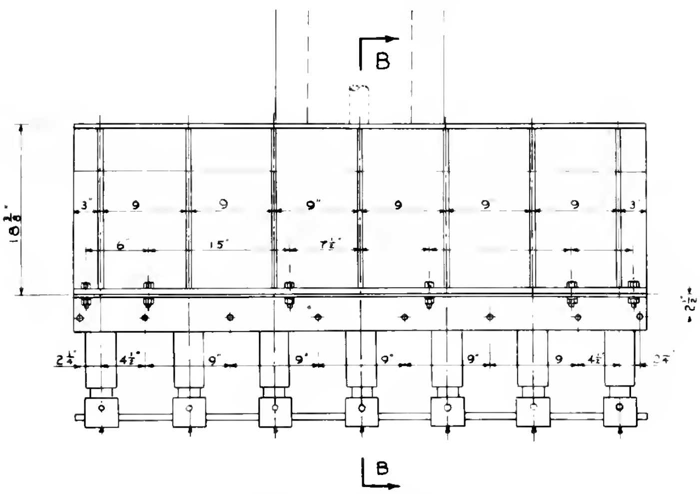
$$a = \text{width of plate} = 13.3125"$$

$$\text{Aspect Ratio} - a/b = \frac{13.3125}{56.25} = 0.237$$

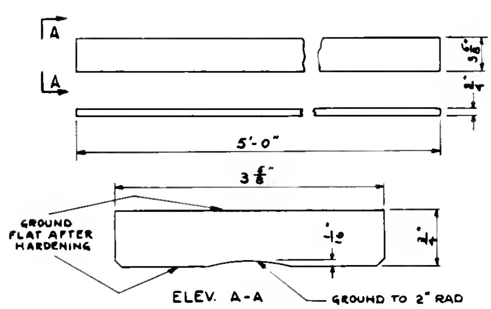
$$\text{Thickness Ratio} - a/t = \frac{13.3125}{t} = 85.4 \quad \text{Plt. \# 1}$$

85.4	Plt. # 1
84.8	Plt. # 2
84.8	Plt. # 3
85.1	Plt. # 4

FIG.
DETAILS OF LOADING ARRANGEMENT

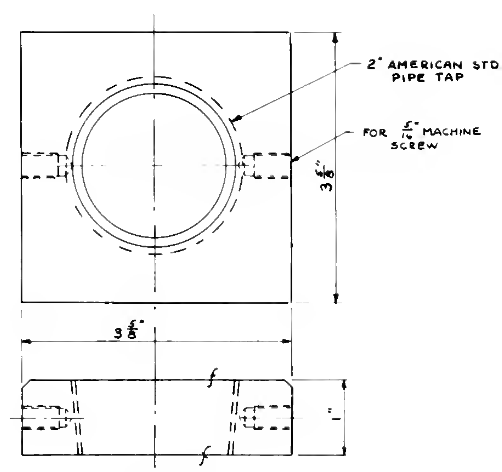


ARRANGEMENT OF UPPER HEAD
LOWER HEAD SIMILAR

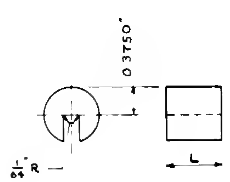


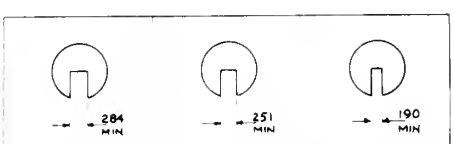
NOTE: CUT FROM $\frac{3}{4}$" STOCK & HARDENED TO ROCKWELL C-50

"B"-EDGE BEARING BAR
MATL: STL AISI C-1045



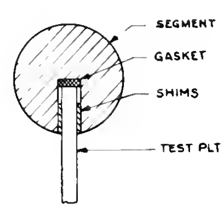
JACK FOOT
MATL: MILD STEEL





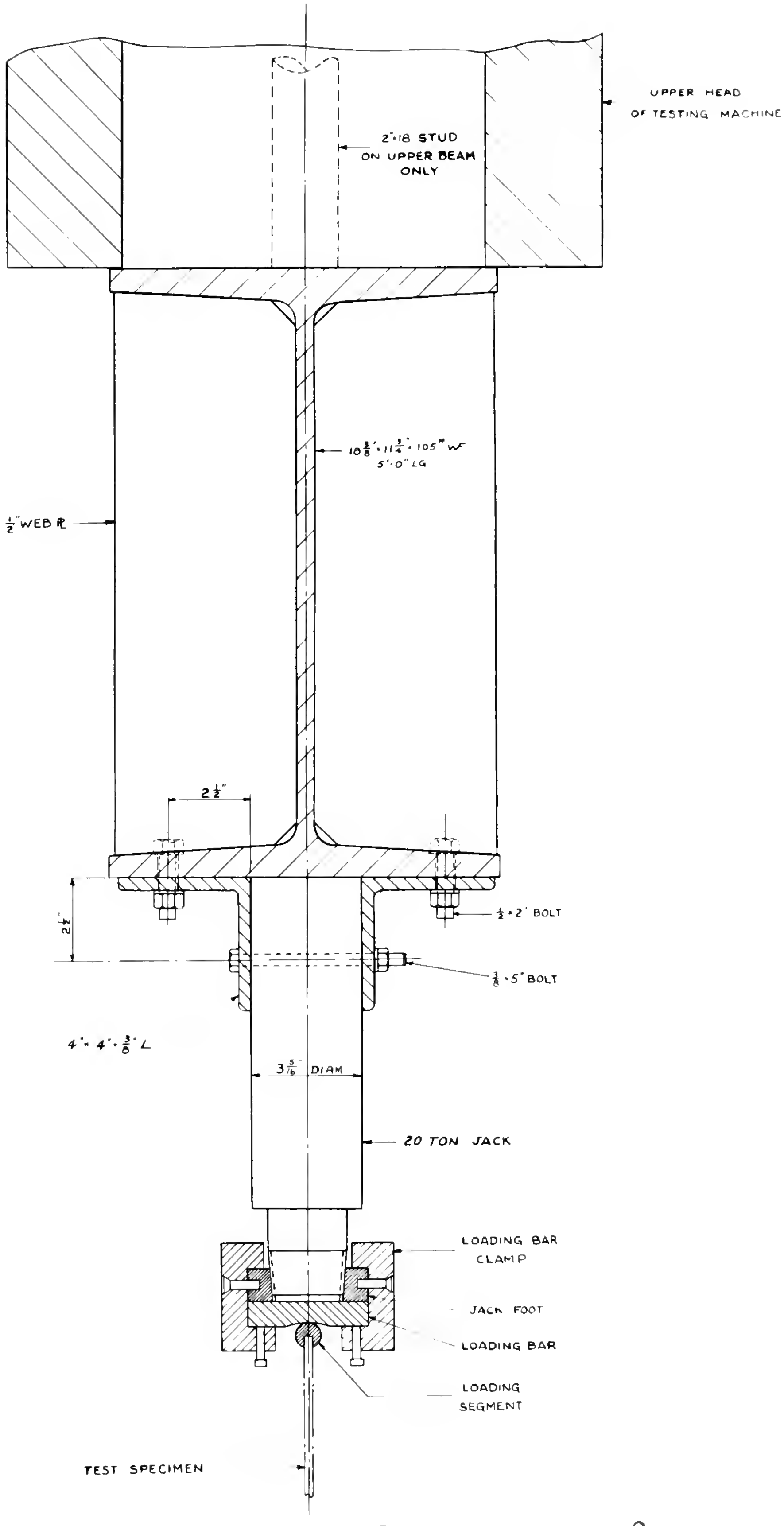
QTY	L	QTY	L	QTY	L
44	0.50	44	0.50	44	0.50
44	0.875	74	0.875	94	0.875

"B"-EDGE BEARING SEGMENTS
MATL: $\frac{3}{4}$" DRILL ROD
MILLED & HARDENED TO ROCKWELL C-50



DETAIL OF SEGMENT
WITH SHIMS & GASKET

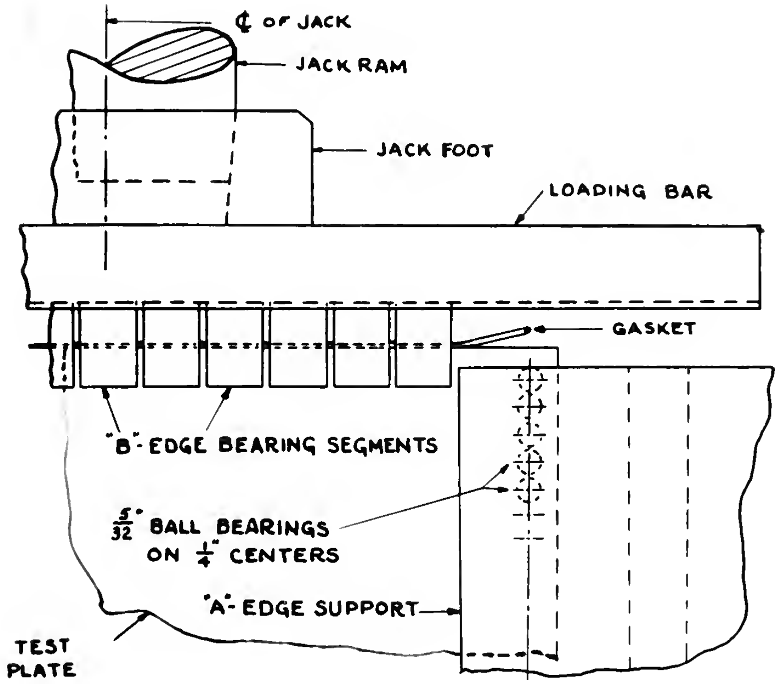
DETAILS OF LOADING ARRANGEMENT



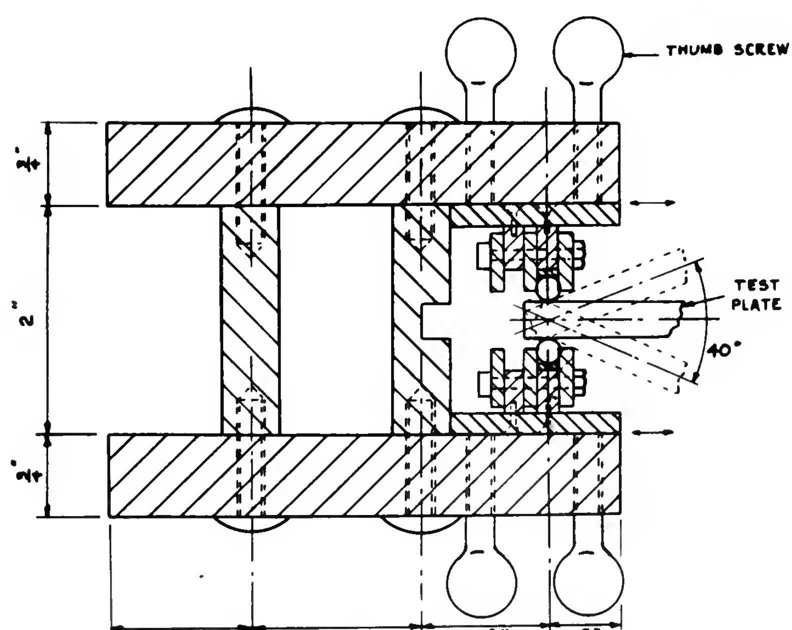
SECTION B-B

MP

FIG.
DETAILS OF LOADING ARRANGEMENT



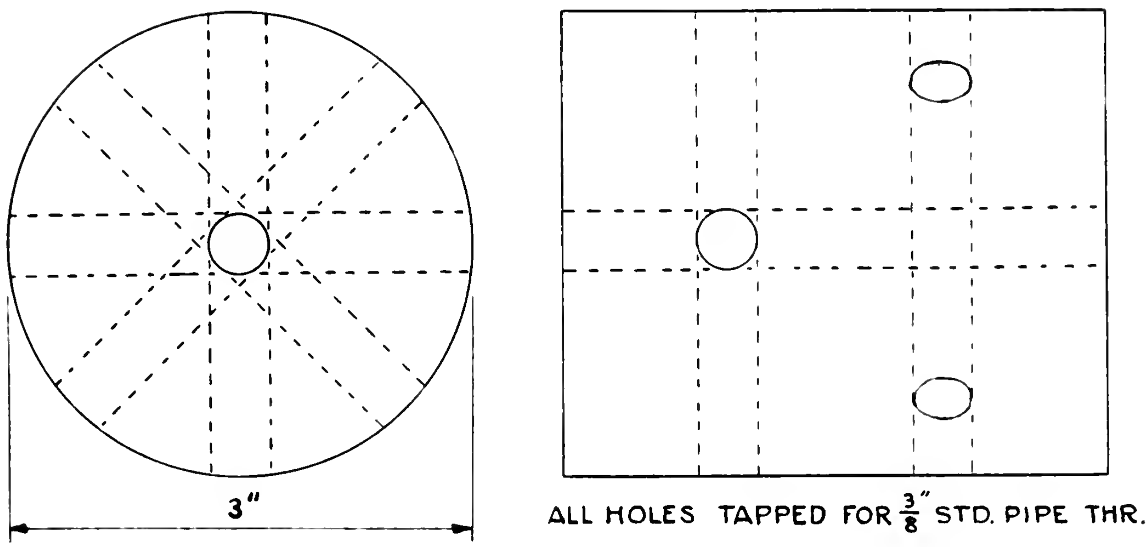
DETAIL AT CORNER OF TEST PLATE
DRAWN TO SCALE
1"



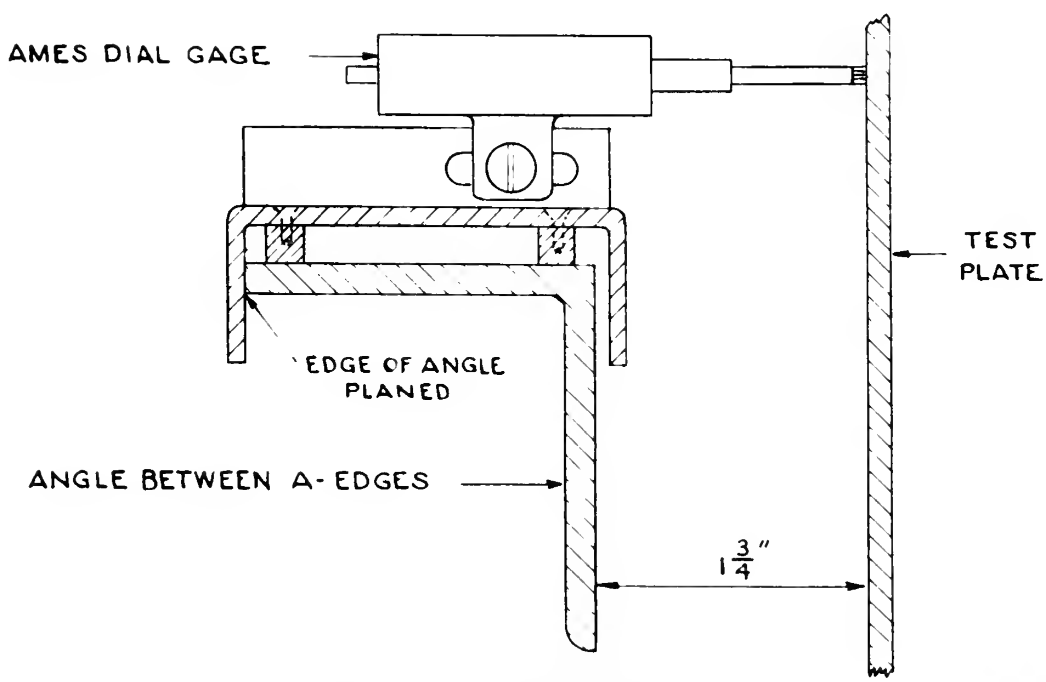
CROSSECTION OF "A"-EDGE SUPPORT

5/7/54 MP

FIG. 30
DETAILS OF LOADING ARRANGEMENT



MANIFOLD



DEFLECTOMETER

5/16/54 M

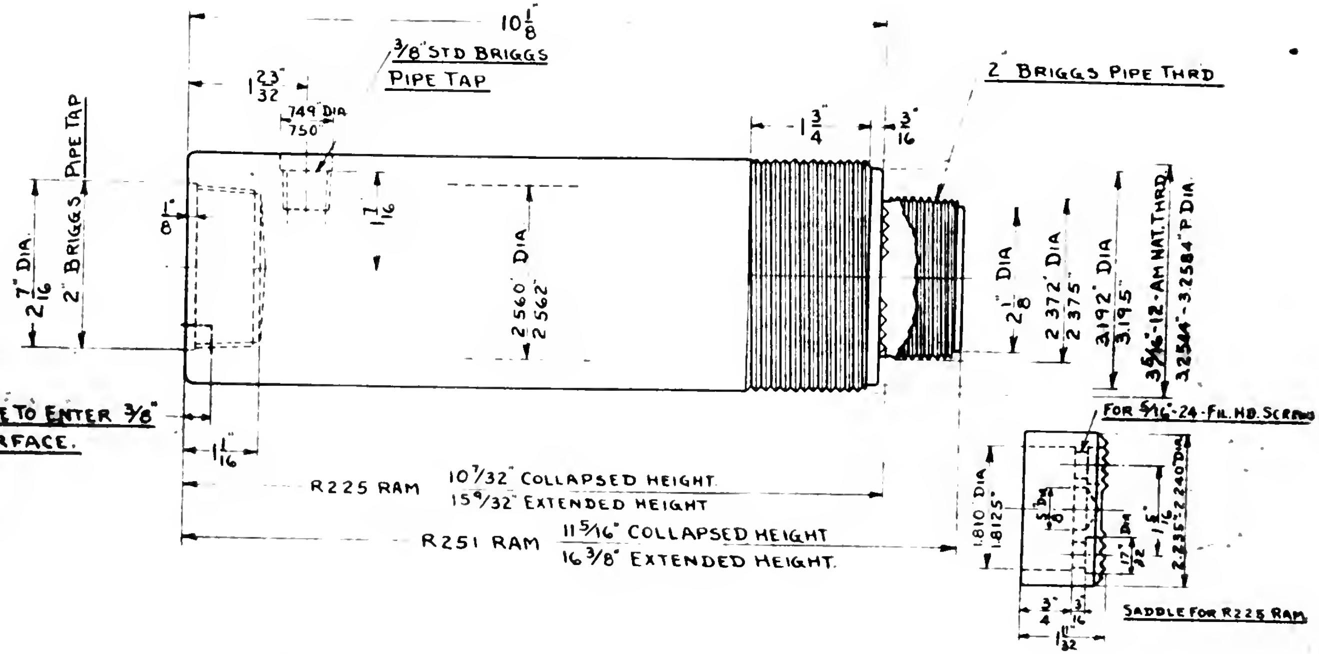
ENG. DEPT.

BLACKHAWK MFG. CO.

MILWAUKEE, WIS.

R225 & R251 PORTO-POWER RAM SPECIFICATIONS.

OCT. 23, 1941.



R225 RAM WEIGHT 20¼ LBS
R251 RAM WEIGHT 21½ LBS
RATED CAPACITY 20 TON
RAM DIA. 2.562"
RAM TRAVEL 5.062"
RAM AREA 5.1572 SQ. IN.
OIL CAPACITY 26.1 CU. IN.

163 STROKES FOR TOTAL TRAVEL OF 5.062"
.031 INCH PER STROKE OF P76 PUMP
INTERNAL PRESSURE AT RATED CAPACITY OF
7750 LBS. PER SQ. INCH.
HANDLE EFFORT ON PUMP 89 LBS.

USE WITH P76 PUMP.

SHEET No.

FIGURE NO. 32

TEST APPARATUS

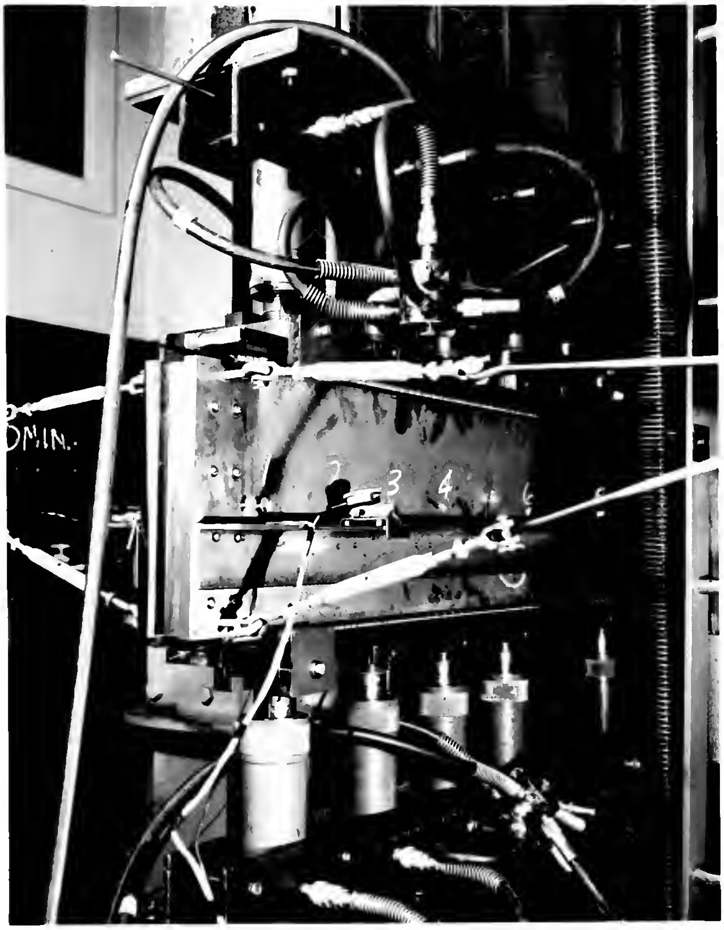


FIGURE NO. 33

TEST APPARATUS

C. Summary of Data and Calculations

Table III A

Evaluation Tests

P = 20,000 lb. ε_{av} = 81.1 in./in.

Test No.	*	ε av	Lowest ε_a	% under	Highest ε_a	% over	Max. %
1	T	85.5	46.5	42.7	153.0	88.6	88.6
	B	78.0	35.0	56.9	152.0	87.5	
4	T	82.2	48.0	40.8	126.5	56.0	56.0
	B	69.7	39.5	51.4	94.5	16.6	
5	T	76.9	51.5	36.6	94.5	16.6	43.9
	B	70.1	45.5	43.9	102	25.8	
6	T	76.1	48	40.8	110.5	36.3	40.8
	B	69.0	47.5	41.5	96.5	19.0	
7	T	69.8	34.5	57.5	97.0	19.6	71.0
	B	71.2	23.5	71.0	105.0	29.5	
13	T	81.7	55.5	31.6	111	36.9	49.3
	B	78.1	47.0	42.1	121	49.3	
14	T	78.0	47.0	42.1	138.5	70.8	70.8
	B	72.6	52	35.9	95.5	17.8	
15	T	66.1	22	73.0	118.5	46.2	74.8
	B	62.7	20.5	74.8	88.0	8.5	
16	T	72.0	52.5	35.3	145.5	79.4	79.4
	B	72.4	26	68.0	112.5	38.8	
17	T	69.3	35	56.9	118.5	46.2	81.5
	B	64.1	15	81.5	120.5	48.6	
18	T	70.9	29	64.3	106	30.7	79.1
	B	70.2	17	79.1	112.5	38.8	

* T = Top of plate ; B = Bottom of plate

C. Summary of Data and Calculations

Table III A

Evaluation Tests

P = 20,000 lb. ε_{av} = 81.2 in./in.

Test No.	*	ε_{av}	Lowest ε_s	% under	Highest ε_s	% over	Max. %
1	T	85.5	46.5	42.7	153.0	88.?	88.6
	B	78.0	35.0	56.9	152.?	87.?	
4	T	82.2	48.0	40.8	120.5	[illegible]	56.0
	B	69.7	39.5	51.4	94.5	[illegible]	
5	T	76.9	51.5	36.6	94.?	[illegible]	43.9
	B	70.1	45.5	43.5	102	[illegible]	
6	T	76.1	48	40.9	110.?	35.?	40.6
	B	69.0	47.5	41.5	96.?	19.?	
7	T	69.8	34.5	57.5	97.0	[illegible]	71.0
	B	71.2	23.5	71.0	105.?	[illegible]	
13	T	81.7	55.5	31.?	111	36.9	49.3
	B	78.1	47.0	42.1	121	49.?	
14	T	78.0	47.0	42.?	13?.5	70.?	70.6
	B	72.6	52	35.9	95.?	[illegible]	
15	T	66.1	22	73.0	12?.5	[illegible]	[illegible]
	B	62.7	20.5	74.2	88.0	[illegible]	
16	T	72.0	52.5	35.3	145.5	79.?	79.?
	B	72.4	26	68.0	122.5	50.?	
17	T	69.3	35	56.9	118.5	[illegible]	[illegible]
	B	64.1	15	[illegible]	120.?	[illegible]	
18	T	70.9	29	6?.?	16?	[illegible]	[illegible]
	B	70.2	17	79.1	132.5	[illegible]	

* T = Top of plate ; B = Bottom of plate

Table III B

Buckling Tests

Plate No.	1	2	3	4
a (in.)	13.313	13.313	13.313	13.313
b (in.)	56.75	56.75	56.75	56.75
t (in.)	0.1535	1.545	1.545	1.540
a/b	0.237	0.237	0.237	0.237
a/t	86.8	86.2	86.2	86.5
b/t	370	367	367	368
d_o (measured) (in.)	0.085	0.057	0.009	0.071
d_o (Donnell) (in.)	0.038	0.059	0.018	0.060
d_o (Southwell) (in.)	0.044	0.061	0.019	0.062
σ ult (exp.) (psi)	4140	3375	3720	3830
σ cr (Theory) (psi)	3780	3830	3830	3810
σ cr ("T.O.K.") (psi)	3730	2620	3210	3190
σ cr (Donnell) (psi)	4640	3530	3580	4120
σ cr (Southwell) (psi)	4520	3540	3635	3940
$\frac{\text{Donnell} - \text{Bleich}}{\text{Bleich}} \times 100$	+ 23.4	-7.83	-6.53	+ 7.61

Table III B

Buckling Tests

Plate No.	1	2	3	4
a (in.)	13.313	13.323	[illegible]	13.313
b (in.)	56.75	56.75	56.75	56.75
t (in.)	0.1535	0.1545	[illegible]	0.1540
a/b	0.237	0.237	[illegible]	0.237
a/t	86.8	86.2	[illegible]	86.5
b/t	370	367	[illegible]	368

δ_0 (measured) (in.)	0.085	0.057	[illegible]	0.071
δ_0 (Donnell) (in.)	0.098	0.079	0.015	0.060
δ_0 (Southwell) (in.)	0.044	[illegible]	0.039	[illegible]

σ ult (exp.) (psi)	4140	3375	[illegible]	3830
σ cr (Theory) (psi)	3760	[illegible]	[illegible]	3810
σ cr ("T.O.K.") (psi)	2730	[illegible]	[illegible]	[illegible]
σ cr (Donnell) (psi)	[illegible]	5530	[illegible]	[illegible]
σ cr (Southwell) (psi)	[illegible]	[illegible]	[illegible]	3940
$\frac{\text{Donnell} - \text{Bleich}}{\text{Bleich}} \times 100$	+ [illegible]	[illegible]	[illegible]	+ [illegible]

D. SAMPLE CALCULATIONS

1. The longitudinal strain in the plate at a given point on its neutral axis was obtained by averaging the strains obtained from a pair of gages mounted on opposite sides of the plate and opposite each other as shown on Fig. 15. All odd numbered gages were on one side of the plate, and all even numbered gages were on the other, the arrangement being such that each odd gage had opposite it the gage with the next highest even number.

$$\varepsilon_a = \frac{\varepsilon_1 + \varepsilon_2}{2}$$

Thus for gage numbers 67 and 68 in Test No. 19 on Plate No. 1, at a load of 20,000 pounds,

$\varepsilon_1 = +181$ microinches/inch

$\varepsilon_2 = -359$ microinches/inch

$\varepsilon_a = \frac{(+181) + (-359)}{2} = -89$ microinches/inch

2. The average longitudinal stress in the plate was obtained by dividing the total applied load by the edge area of the plate.

$$\sigma_{av} = \frac{P}{b'xt}$$

For Test No. 19 on Plate No. 1 at a load of 20,000 pounds,

$P = 20{,}000$ lbs.

$b' = 56.75$ in.

$t = 0.1535$ in.

$\sigma_{av} = \frac{20{,}000}{(56.75)(0.1535)} = 2295$ psi

For the same test, at the critical load,

$P_{cr} = 40{,}400$ lbs. (Donnell)

$\sigma_{av} = \frac{40{,}400}{(56.75)(0.1535)} = 4640$ psi

B. SAMPLE CALCULATIONS

1. The longitudinal strain in the plate at a given point on the neutral axis was obtained by averaging the strains obtained from a pair of gages mounted on opposite sides of the plate and opposite each other as shown on Fig. 15. All odd numbered gages were on one face of the plate, and all even numbered gages were on the other, the arrangement being such that each odd gage had opposite it the gage with the next highest even number.

$$\epsilon_a = \frac{\epsilon_1+\epsilon_2}{2}$$

Thus for gage numbers 67 and 68 in Test No. 19 on Plate No. 1, at a load of 20,000 pounds,

$\epsilon_1 = +181$ microinches/inch

$\epsilon_2 = -359$ microinches/inch

$$\epsilon_a = \frac{(+181)+(-359)}{2} = -89 \text{ microinches/inch}$$

2. The average longitudinal stress in the plate was obtained by dividing the total applied load by the edge area of the plate.

$$\sigma_{av} = \frac{P}{b't}$$

For Test No. 19 on Plate No. 1 at a load of 20,000 pounds,

P = 20,000 lbs.

b' = 56.75 in.

t = 0.1535 in.

$$\sigma_{av} = \frac{20,000}{(56.75)(0.1535)} = \text{[illegible] psi}$$

For the same test, at the critical load

P_{cr} = 40,400 lbs. (Connell)

$$\sigma_{av} = \frac{40,400}{(56.75)(0.1535)} = \text{[illegible] psi}$$

3. <u>Average strain</u> was calculated using the stresses obtained above and the value of E obtained from Appendix F.

$$\mathcal{E}_{av} = \frac{\sigma_{av}}{E}$$

$$E = 28{,}300{,}000 \text{ psi}$$

At 20,000 pounds,

$$\mathcal{E}_{av} = \frac{2295}{28{,}300{,}000} = 81.1 \text{ microinches/inch}$$

A different value for $\mathcal{E}_{av}$ is obtained if the plot of axial strain (ε_a) is integrated by the trapezoidal rule along the line of gages, numbers 7, 8 through 35, 36. (A value of 81.1 will be assumed for the axial strain at gages 35 and 36, since no readings were obtained at these gages for this test.)

$$\mathcal{E}_{av} = \frac{1}{14}\left[\frac{1}{2}\varepsilon_{a(7,8)} + \sum_{(9,10)}^{(33,34)} \varepsilon_a + \frac{1}{2}\varepsilon_{a(35,36)}\right]$$

$$\mathcal{E}_{av} = \frac{1}{14}(1046.6) = 74.8 \text{ microinches/inch}$$

4. <u>Percentage of maximum deviation</u> of axial strain from average strain as computed from the total load.

$$\% = \frac{\varepsilon_a - \mathcal{E}_{av}}{\mathcal{E}_{av}} \times 100$$

$$\mathcal{E}_{av} = \frac{P}{Ebt} = 81.1 \text{ microinches/inch}$$

$$\mathcal{E}_a \text{ max} = 116 \text{ microinches/inch}$$

$$\mathcal{E}_a \text{ min} = 50 \text{ microinches/inch}$$

$$\text{Max \% over} = \frac{116-81.1}{81.1} \times 100 = 43.0\% \text{ over}$$

$$\text{Max \% under} = \frac{50-81.1}{81.1} \times 100 = 38.4\% \text{ over}$$

5. <u>Strain difference</u> (proportional to lateral deflection) was obtained by subtracting readings of opposite gages. Arbitrarily this was taken as

$$\varepsilon_2 - \varepsilon_1 = \varphi(d)$$

For Test No. 19 on Plate No. 1, using the values in (1) above,

$$\varepsilon_2 - \varepsilon_1 = -359 - (+181) = -540 \text{ microinches/inch}$$

6. <u>Critical load (Top-of-the-Knee Method).</u> To obtain this value no calculations other than those given above for σ_{av} and $(\varepsilon_2 - \varepsilon_1)$ were necessary. After plotting σ_{av} vs. $(\varepsilon_2 - \varepsilon_1)$ the tangents at the extremities of the curve were drawn. The intersection was taken as the buckling point.

$$\sigma_{cr} = 3730 \text{ psi}$$

7. <u>Critical load (Southwell's Method).</u>

$$\varepsilon_2 - \varepsilon_1 = \varphi(d)$$

$$\frac{\varepsilon_2 - \varepsilon_1}{P} = \frac{\varphi(d)}{\text{Load}}$$

For gages number 67 and 68 in Test No. 19, Plate No. 1 at a load of 20,000 pounds,

$$P = 20{,}000 \text{ lbs.}$$

$$\varepsilon_2 - \varepsilon_1 = -540 \text{ microinches/inch (from above)}$$

$$\frac{\varepsilon_2 - \varepsilon_1}{P} = \frac{540}{20{,}000} = 0.0270$$

A plot of $\frac{\varepsilon_2 - \varepsilon_1}{P}$ vs. $\varepsilon_2 - \varepsilon_1$ gives a straight line with a slope equal to the critical load.

5. Strain difference (proportional to lateral deflection) was obtained by subtracting readings of opposite gages. Arbitrarily this was taken as

$$\epsilon_2 - \epsilon_1 = \varphi(d)$$

For Test No. 19 on Plate No. 1, using the values in 1. above,

$$\epsilon_2 - \epsilon_1 = -359 - (+181) = -540 \text{ microinches/inch}$$

6. Critical Load (Top-of-the-Knee Method). To obtain this value no calculations other than those given above for σ_{av} and $(\epsilon_2 - \epsilon_1)$ were necessary. After plotting σ_{av} vs. $(\epsilon_2 - \epsilon_1)$ the tangents at the extremities of the curve were drawn. The intersection was taken as the buckling point.

$$\sigma_{cr} = 3720 \text{ psi}$$

7. Critical Load (Southwell's Method).

$$\epsilon_2 - \epsilon_1 = \varphi(d)$$

$$\frac{\epsilon_2 - \epsilon_1}{P} = \frac{\varphi(d)}{\text{Load}}$$

For gages number 67 and 68 in Test No. 19, Plate No. 1 at a load of 20,000 pounds,

$$P = 20{,}000 \text{ lbs.}$$

$$\epsilon_2 - \epsilon_1 = -540 \text{ microinches/inch (from above)}$$

$$\frac{\epsilon_2 - \epsilon_1}{P} = \frac{540}{20{,}000} = 0.0270$$

A plot of $\frac{\epsilon_2 - \epsilon_1}{P}$ vs. $\epsilon_2 - \epsilon_1$ gives a straight line with a slope equal to the critical load.

$$P_{cr} = \frac{\partial[(\varepsilon_2 - \varepsilon_1)]}{\partial[(\varepsilon_2 - \varepsilon_1)/P]}$$

$$P_{cr} = 39{,}400 \text{ lbs.}$$

$$\sigma_{cr} = \frac{P_{cr}}{b' x t} = 4520 \text{ psi}$$

The intercept of the straight line with the $(\varepsilon_2 - \varepsilon_1)$ axis represents an equivalent initial deflection of the plate.

$$(\varepsilon_2 - \varepsilon_1)_o = 355 \text{ microinches/inch}$$

$$w = d_o \sin \frac{\pi x}{L}$$

$$M_x = -D \frac{\partial^2 w}{\partial x^2} = D d_o \frac{\pi^2}{L^2} \sin \frac{\pi x}{L}$$

$$M_x = D d_o \frac{\pi^2}{L^2} \quad \text{when } x = \frac{L}{2}$$

$$\sigma_{max} = \frac{M_x t}{2 I} = \frac{E(\varepsilon_2 - \varepsilon_1)}{2}$$

$$M_x = \frac{EI(\varepsilon_2 - \varepsilon_1)}{t}$$

$$D = \frac{Et^3}{12(1-\nu^2)} = \frac{EI}{1-\nu^2}$$

$$\frac{EI}{1-\nu^2} \frac{\pi^2}{L^2} d_o = \frac{EI(\varepsilon_2 - \varepsilon_1)}{t}$$

$$d_o = \frac{(1-\nu^2) L^2}{\pi^2 t} (\varepsilon_2 - \varepsilon_1)$$

$\nu = 0.3$

$L = 13.313$ in.

$t = 0.1535$ in.

$(\varepsilon_2 - \varepsilon_1) = 355 \times 10^{-6}$ in./in (from above)

$$d_o = \frac{0.91}{\pi^2} \frac{(13.313)^2}{0.1535} (355 \times 10^{-6}) = 0.0378 \text{ in.}$$

$$P_{cr} = \frac{\delta[(\epsilon_2 - \epsilon_1)]}{\delta[(\epsilon_2 - \epsilon_1)/P]}$$

$$P_{cr} = 79,400 \text{ lbs.}$$

$$\sigma_{cr} = \frac{P_{cr}}{b't} = 15520 \text{ psi}$$

The intercept of the straight line with the $(\epsilon_2 - \epsilon_1)$ axis represents an equivalent initial deflection of the plate.

$$(\epsilon_2 - \epsilon_1)_o = 355 \text{ microinches/inch}$$

$$w = \delta_o \sin \frac{\pi x}{L}$$

$$M_x = -D \frac{\partial^2 w}{\partial x^2} = D\delta_o \frac{\pi^2}{L^2} \sin \frac{\pi x}{L}$$

$$M_x = D\delta_o \frac{\pi^2}{L^2} \quad \text{when } x = \frac{L}{2}$$

$$\sigma_{max} = \frac{M_x t}{2I} = \frac{E(\epsilon_2 - \epsilon_1)}{2}$$

$$M_x = \frac{EI(\epsilon_2 - \epsilon_1)}{t}$$

$$D = \frac{Et^3}{12(1-\nu^2)} = \frac{EI}{1-\nu^2}$$

$$\frac{EI}{1-\nu^2} \delta_o \frac{\pi^2}{L^2} = \frac{EI(\epsilon_2 - \epsilon_1)}{t}$$

$$\delta_o = \frac{(1-\nu^2) L^2}{\pi^2 t} (\epsilon_2 - \epsilon_1)$$

$$\nu = 0.3$$

$$L = 13.313 \text{ in.}$$

$$t = 0.1535 \text{ in.}$$

$(\epsilon_2 - \epsilon_1) = 355 \times 10^{-6}$ in./in (from above)

$$\delta_o = \frac{0.91}{\pi^2} \frac{(13.313)^2}{0.1535} (355 \times 10^{-6}) = \text{[illegible]}$$

8. <u>Critical load (Donnell's Method).</u> Donnell's type of plot determines the critical load directly as the intercept on the P-axis of the plot of P vs. $P/(\varepsilon_2 - \varepsilon_1)$.

For Test No. 19, Plate No. 1,

$P_{cr} = 40{,}400$ lbs.

$$\sigma_{cr} = \frac{P_{cr}}{b't} = 4640 \text{ psi}$$

The slope of the plot is the equivalent initial deflection

$$(\varepsilon_2 - \varepsilon_1)_o = \frac{d\,[P]}{d\,[P/(\varepsilon_2 - \varepsilon_1)]} = 415 \text{ microinches/inch}$$

$d_o = 0.0442$ in.

9. <u>Theoretical Critical Stress</u>

a. $$\frac{\sigma_{cr}}{\tau} = \frac{\pi^2 E}{12(1-\nu^2)}\left(\frac{t}{a}\right)^2 \bar{k}$$

$E = 28{,}300{,}000$ lbs/in.^{2}

$\nu = 0.3$ (assumed)

$a = 13.313$ in.

$b = 56.25$ in.

$\frac{a}{b} = 0.237$

$\bar{k} = 1.113$ (Table 40, p. 436)

$\tau = 1.000$

$\sigma_{cr} = 160{,}400\ t^2$

Plate No.	t (in.)	σ_{cr} (psi)
1	.1535	3780
2	.1545	3830
3	.1545	3830
4	.1540	3810

8. Critical Load (Donnell's Method). Donnell's type of plot determines the critical load directly as the intercept on the P-axis of the plot of P vs. $P/(\epsilon_2 - \epsilon_1)$.

For Test No. 19, Plate No. 2,

$P_{cr} = 40,400$ lbs.

$$\sigma_{cr} = \frac{P_{cr}}{2bt} = 4640 \text{ psi}$$

The slope of the plot is the equivalent initial deflection

$$(\epsilon_2 - \epsilon_1)_0 = \frac{d[\epsilon]}{d[P/(\epsilon_2 - \epsilon_1)]} = 425 \text{ microinches}$$

$d_0 = 0.0442$ in.

9. Theoretical Critical Stress

a. $\frac{\sigma_{cr}}{\tau} = \frac{\pi^2 E}{12(1-\nu^2)} \left(\frac{t}{a}\right)^2 k$

$E = 28,300,000$ lbs/in.^{2}

$\nu = 0.3$ (assumed)

$a = 13.313$ in.

$b = 56.25$ in.

$\frac{a}{b} = 0.237$

$K = 1.113$ (Table 40, p. 434)

$\tau = 1.000$

$\sigma_{cr} = 110,400\ t^2$

Plate No.	t, in.	σ_{cr}, psi
1	[illegible]	[illegible]
2	[illegible]	[illegible]
3	[illegible]	[illegible]
4	[illegible]	[illegible]

b. $$\sigma_{cr} = \frac{\pi^2 E}{12(1-\nu^2)} \left(\frac{t}{b}\right)^2 k \qquad (3)$$

where $E = 28{,}300$ lb./in.^{2}

$\nu = 0.3$

$a = 13.313$ in.

$b = 56.25$ in.

$\alpha = \frac{a}{b} = 0.237$

$k = \left(\frac{\alpha}{n} + \frac{n}{\alpha}\right)^2$ for plates simply supported on all four edges

$n = 1$

$$k = \left(\frac{0.237}{1} + \frac{1}{0.237}\right)^2 = 19.90$$

$\sigma_{cr} = 161{,}000\ t^2$

Plate No.	σ_{cr} (psi)
1	3790
2	3840
3	3840
4	3820

10. Modulus of Elasticity

For specimen number 1:

$\ell = 2.547$ in. Huggenberger G. F. (#1) = 1051

$w = 0.8435$ in. Huggenberger G. F. (#2) = 1040

$t = 0.1572$ in.

At P = 0: $R_1 = 1.55$

$R_2 = 1.51$

At P = 2160 lbs.: $R_1 = 1.09$

$R_2 = 0.86$

b. $\sigma_{cr} = \frac{\pi^2 E}{12(1-\nu^2)} \, k \left(\frac{t}{b}\right)^2$ (3)

where $E = 28,300$ lb./in.^{2}

$\nu = 0.3$

$a = 13.313$ in.

$b = 56.25$ in.

$\alpha = \frac{a}{b} = 0.237$

$k = \left(\frac{m}{\alpha} + \frac{\alpha}{m}\right)^2$ for plates simply supported on all four edges

$m = 1$

$k = \left(\frac{0.237}{1} + \frac{1}{0.237}\right)^2 = 19.90$

$\sigma_{cr} = 161,000 \, t^2$

Plate No.	σ_{cr} (psi)
1	3790
2	3840
3	3840
4	3820

10. Modulus of Elasticity

For specimen number 1:

$l = 2.547$ in. Huggenberger G. 1. ([illegible]) = [illegible]

$w = 0.8435$ in. Huggenberger G. 2. ([illegible]) = [illegible]

$t = 0.1579$ in.

At P = 0: $R_1 = 1.55$

$R_2 = 1.51$

At P = 2160 lbs.: $R_1 = 1.09$

$R_2 = 0.84$

$$\sigma = \frac{P}{b't} = \frac{2.16}{(0.8435)(0.1572)} = 16.28 \text{ k.s.i.}$$

$$\varepsilon_1 = \frac{1.55 - 1.09}{1051} = 437.5 \text{ microinches/inch}$$

$$\varepsilon_2 = \frac{1.51 - 0.86}{1040} = 625 \text{ microinches/inch}$$

$$\varepsilon = \frac{\varepsilon_1 + \varepsilon_2}{2} = 531 \text{ microinches/inch}$$

From a composite plot of all three specimens

$$E = \frac{\Delta\sigma}{\Delta\varepsilon} = \frac{40.0}{1415\times10^{6}} = 28{,}300 \text{ k.s.i.}$$

E. SUPPLEMENTARY DISCUSSION

1. Comparison of Methods for Determining σ_{cr}

The "top-of-the-knee" method gives values which are lower than the theoretical buckling stresses and are also the lowest given by any experimental analysis except for the strain-reversal method which was not usable in this case. This is in line with previous experience that this method gives low values whenever there is appreciable initial unfairness. To obtain better results it is necessary to apply the method to a series of plots for plates of the same size but having different initial deflections. If a line is then drawn through the tangent intersections and extrapolated to the σ-axis, the intercept represents the critical stress for a plate of zero initial deflection. This method was not usable for the series of plates tested in this work since there appeared to be no well-defined line through the points. A further difficulty in using the "top-of-the-knee" method is that a large number of points must be taken around the buckling point, and even then the upper tangent may be somewhat difficult to draw.

This same difficulty is inherent in Yoshiki's method, although the difficulty of drawing the tangent is reduced if a sufficient number of points are taken, and the knee appears sharper.

The Southwell method [48] and Donnell's variation [18] are based upon column theory rather than plate theory. However, these methods can be applied to some plate situations, especially when the aspect ratio is small and the $\frac{a}{t}$ ratio is large as is true for this case. For other plates these methods may be invalid, but the fact that the plots are straight lines for the present tests is strong evidence that the results are good.

2. SUPPLEMENTARY DISCUSSION

1. Comparison of Methods for Determining σ_{cr}

The "top-of-the-knee" method gives values which are lower than the theoretical buckling stresses and are also the lowest given by any experimental analysis except for the strain-reversal method which was not usable in this case. This is in line with previous experience that this method gives low values whenever there is appreciable initial unflatness. To obtain better results it is necessary to apply the method to a series of plots for plates of the same size but having different initial deflections. If a line is then drawn through the tangent intersections and extrapolated to the σ-axis, the intercept represents the critical stress for a plate of zero initial deflection. This method was not usable for the series of plates tested in this work since there appeared to be no well-defined line through the points. A further difficulty in using the "top-of-the-knee" method is that a large number of points must be taken around the buckling point, and even then the upper tangent may be somewhat difficult to draw.

This same difficulty is inherent in Fowler's method, although the difficulty of drawing the tangent is reduced if a sufficient number of points are taken, and the knee appears sharper.

The Southwell method [48] and Donnell's variation [2] are based upon column theory rather than plate theory. However, these methods can be applied to some plate situations, especially when the deflections are small and the $\frac{b}{t}$ ratio is large as it was for this case. For thick plates these methods may be invalid, but the fact that the plots are straight lines for the present tests is strong evidence that the results are good.

In any event, either method may be helpful in that additional points can be taken from the straight line plot and used to more fully define the σ-d curve when the "top-of-the-knee" or Yoshiki's method is used. Donnell's method seems somewhat preferable when it can be used since the plotted points are evenly spaced (for even increments of load) and since the critical load can be read directly as the intercept on the P-axis.

2. Deflection Measurements

During the tests numerous lateral deflection measurements were made along the plate, using the planed edge of the angle connecting the "a"-edge supports as the reference line. These proved to be practically useless, since the plate adjusted itself in the early stages of loading, and the deflections for the lower loads did not reflect bending in the plate. Attempts to use these values when making buckling analyses generally failed.

A far better measurement of the lateral deflections in the plate was the difference of the strains in opposite pairs of gages. These differences are directly proportional to the moment, and when the deformation of the plate is sinusoidal, they are proportional to the lateral deflection. These values are free of the effects of any lateral movement (without bending) which might occur as the plate adjusts itself to fit in the grooves in the loading bar. Furthermore, these readings do not reflect residual stresses which may be present in the plate.

Initial deflections were measured at various points on the four plates prior to placing them in the test apparatus. This was accomplished by laying a straight edge across the short dimension of the plate, and measuring the largest unfairness with a feeler gage. A plot

In any event, either method may be helpful in that sufficient [illegible] can be taken from the straight line plot [illegible] define the σ-d curve when the "tangent-modulus" [illegible] is used. Donnell's method seems somewhat preferable [illegible] used since the plotted points are evenly spaced [illegible] (for equal increments of load) and shows the critical load [illegible] directly as the intercept on the P-axis.

2. Deflection Measurements

During the tests numerous lateral deflection measurements were made along the plates, using the planed edge of the angle containing the "V"-edge supports as the reference line. These proved to be practically useless, since the plate adjusted itself in the early stages of loading, and the deflections for the lower loads thus reflect this early [illegible] in the plate. Attempts to use these values when making [illegible] analyses generally failed.

A far better measurement of the lateral deflections in the plate was the difference of the strains on opposite [illegible] These differences are directly proportional to the [illegible] formation of the plate in [illegible] They are [illegible] of the lateral deflection. These values are free of the effects of any [illegible] movement (without bending) which might occur as [illegible] self to fit in the grooves in the loading [illegible]. Furthermore, [illegible] readings do not reflect residual strains [illegible] plate.

Initial deflections were measured at various [illegible] plates prior to placing them in the test apparatus. [illegible] plished by laying a straight edge across the [illegible] plate, and measuring the largest [illegible] of the unfairness along the plate's horizontal centerline did not define any simple curve such as a sine curve. Thus it was impossible to consider these measurements as anything other than an indication of the approximate magnitude of equivalent unfairness. The extrapolated values of d_o taken from the Donnell and Southwell plots compare fairly well with these measured values except for the case of Plate No. 1. (See Table IIIB). In the case of Plate No. 1, the measured values were especially dubious since it was necessary to block the plate into place prior to loading in order to make the long edges approximately straight. The extremely complex initial curvature affected not only the equivalent initial unfairness but apparently made the plate more resistant to buckling, probably since more energy was required to produce the final sinusoidal configuration.

3. Calibration of the Testing Machine

During most of the tests a 0-1000 lb. gage was connected to the upper manifold and the pressure recorded along with the other data. The gage was calibrated at several points and the results plotted. (See Fig. 35) To obtain an equivalent gage pressure as indicated by the testing machine gages, the total indicated load was divided by the total jack ram area of one set of jacks. As can be seen from Fig. 35 the difference at 20,000 lbs. indicated load on the testing machine, and integrated jack loading is about 615 lbs. Extrapolated to 30,000 lbs., the approximate range of ultimate plate loads, this difference becomes about 925 lbs.

If we analyze a free body diagram of a jack it is possible to draw some conclusions about this observed difference in readings.

of the unfairness along the plate's horizontal centerline did not define any simple curve such as a sine curve. Thus it was impossible to consider these measurements as anything other than an indication of the approximate magnitude of equivalent unfairness. The extrapolated values of d_o taken from the Donnell and Southwell plots compare fairly well with those measured values except for the case of Plate No. 1. (See Table III). In the case of Plate No. 1, the measured values were especially dubious since it was necessary to block the plate into place prior to loading in order to make the long edges approximately straight. The extremely complex initial curvature affected not only the equivalent initial unfairness but apparently made the plate more resistant to buckling, probably since more energy was required to produce the final sinusoidal configuration.

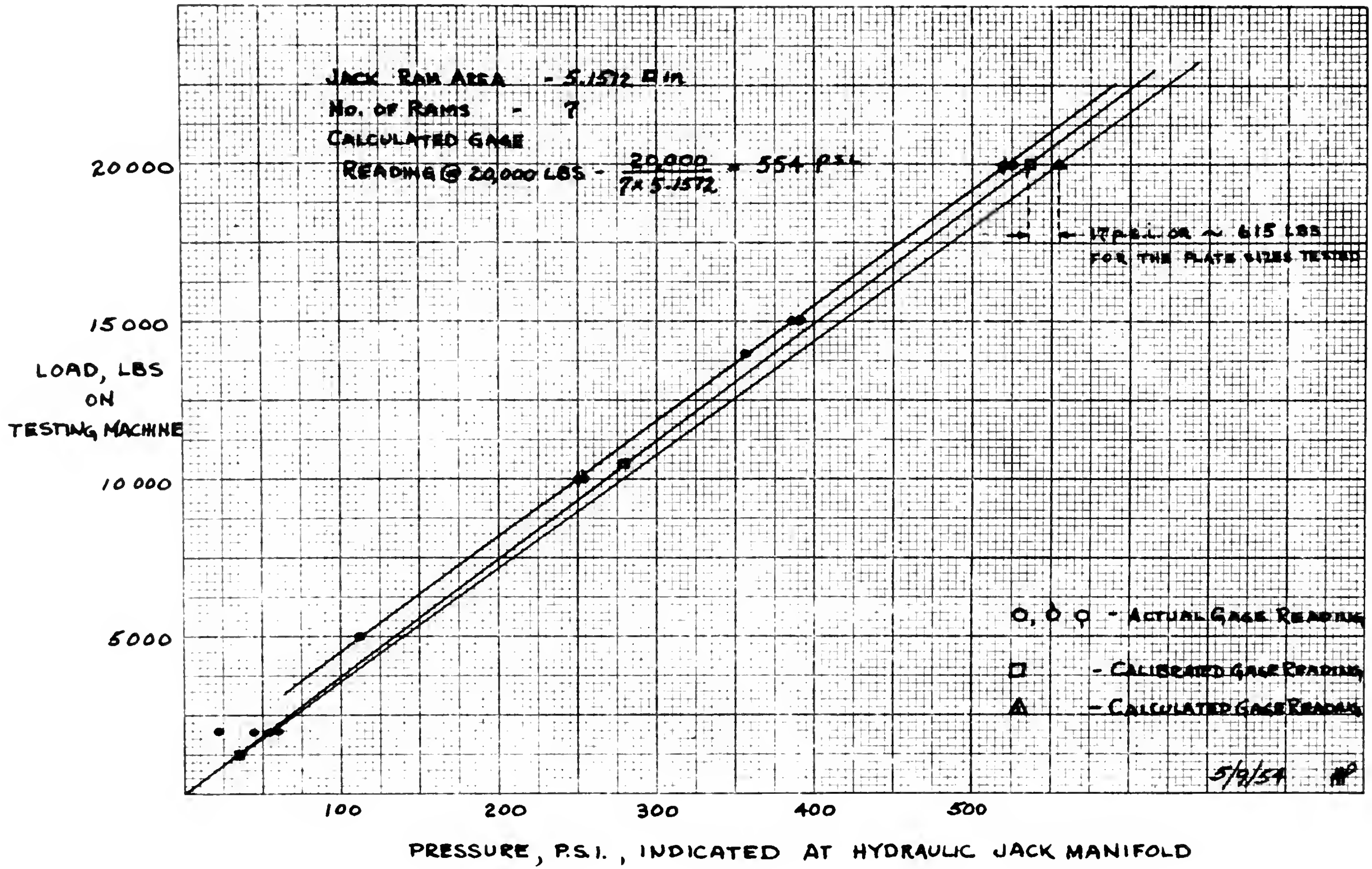
3. <u>Calibration of the Testing Machine</u>

During most of the tests a 0-100K lb. gage was connected to the upper manifold and the pressure recorded along with the other data. The gage was calibrated at several points and the results plotted. (See Fig. 35) To obtain an equivalent gage pressure as indicated by the testing machine gages, the total indicated load was divided by the total jack ram area of one set of jacks. As can be seen from Fig. 35 the difference at 20,000 lbs. indicated load on the testing machine, and integrated jack loading is about 645 lbs. Extrapolated to 70,000 lbs., the approximate range of ultimate plate loads, this difference becomes about 925 lbs.

If we analyze a free body diagram of a jack it is possible to draw some conclusions about this observed difference in loadings.

FIG. 35

CALIBRATION OF TESTING MACHINE



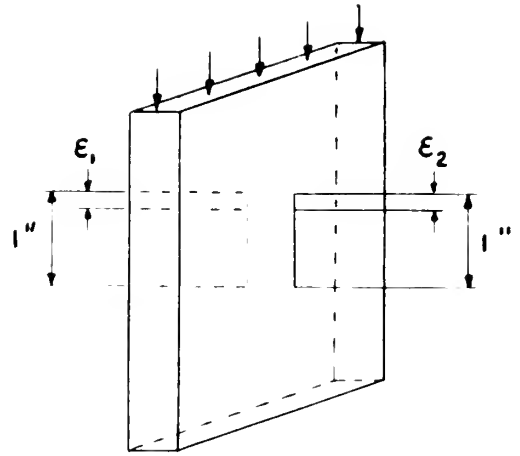
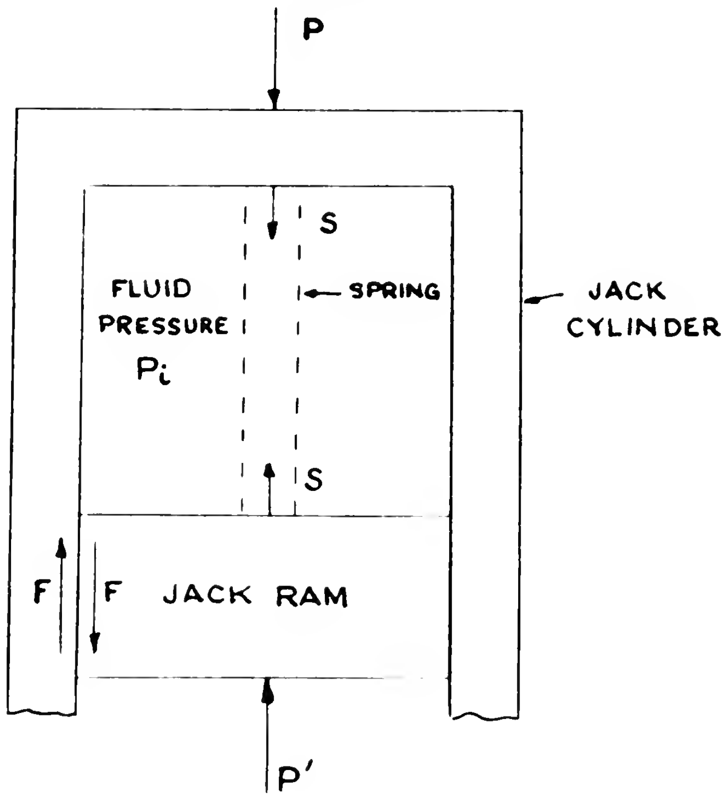


FIG. 36



FREE BODY DIAGRAM
FORCES ON A HYDRAULIC
JACK

From Fig. 36 we can write the following equations:

$$P = P_i A_j - S + F \quad (12)$$

$$P' = P_i A_j - S + F \quad (13)$$

where

P = total load on I-beam, (lbs.)

P' = total load on edge of test plate, (lbs.)

P_i = jack internal hydraulic pressure, (psi)

A_j = total piston area of seven jacks, (sq. in.)

S = total return spring force of seven jacks, (lbs.)

F = total friction force between jack rams and jack cylinders (lbs.)

Under the conditions which prevailed when gage readings were taken the rams could move only inward since the hand pump was not in use. Hence, the direction of the friction force, F, is known.

From equations (12) and (13) we have:

$$P = P' \quad \text{and}$$

$$P - P_i A_j = F - S$$

Unfortunately the pressure gage attached to the jack manifold was not sensitive enough at or near zero readings to give a value for S at different jack extensions.

The results indicate however that if the testing machine gage is a true measure of the load, P,

$$P > P_i A_j \quad \text{or}$$

$$P - P_i A_j > 0 \quad \text{hence}$$

$$F - S > 0 \quad \text{or}$$

$$F > S$$

This result is absurd when $P_i \approx 0$ since the spring force does indeed overcome jack friction and return the rams to the initial position.

From Fig. 36 we can write the following equations:

$$P = P_J A_J - S + F \qquad (12)$$

$$P' = P_J A_J - S + F \qquad (13)$$

where

P = total load on I-beam, (lbs.)

P' = total load on edge of test plate, (lbs.)

P_J = jack internal hydraulic pressure, (psi)

A_J = total piston area of seven jacks, (sq. in.)

S = total return spring force of seven jacks, (lbs.)

F = total friction force between jack rams and jack cylinders (lbs.)

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$$F > S$$

This result is absurd when $P_J \approx 0$ since the spring force does indeed overcome jack friction and return the rams to the initial position.

The validity of this inequality when P_1 is large, cannot be judged accurately from the known data. However, the analysis shows that the observations of Fig. 35 are possible, i.e., that the testing machine gage readings may be correct, but the results also cast reasonable doubt on this assumption. On this basis the authors requested that the gages of machine No. 105 be calibrated at least at one load, 20,000 lbs. It was found that the last calibration test was performed in April, 1950. Unfortunately the calibration was not completed in time for inclusion in this thesis but such a confirmation should be made prior to future buckling tests.

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F. DETERMINATION OF YOUNG'S MODULUS

To determine the modulus of elasticity of the test specimens in compression, a compression jig using the single plate method described and built by Polychrone [40] was used.

Briefly the jig consists of a frame which holds the guide plates and specimen in place and contains a subpress in the form of a plunger which bears on the specimen. The steel guide plates are milled with triangular grooves which run the length of the specimen. The grooves opposite each other in the guide plates are staggered so that lateral extension of the specimen is permitted when a load is applied. Two Huggenberger strain gages of one inch base length are attached to opposite edges of the specimen through apertures in the frame. Guide plate pressure and centering is accomplished by adjusting two 3/8" knurled screws until they are just hand tight.

The specimens were rough cut from the same plate and in the same direction as were the test plates. Test plates and compression specimens received the same surface preparation such as sand- or shot-blasting or pickling.

Three specimens were prepared and milled to the final dimensions. The thickness was determined in the same manner as that described for the test plates. All specimens were of the same length and width, 0.8435" x 2.547" which conform to the ASTM standards of a 1 to 3 proportion. (Designation E9-33T) The results of the three tests are given in Fig. 37. There are several possible sources of error in this procedure. The grooved guide plates do not permit complete freedom of lateral extension. This has effect primarily on the determination of the

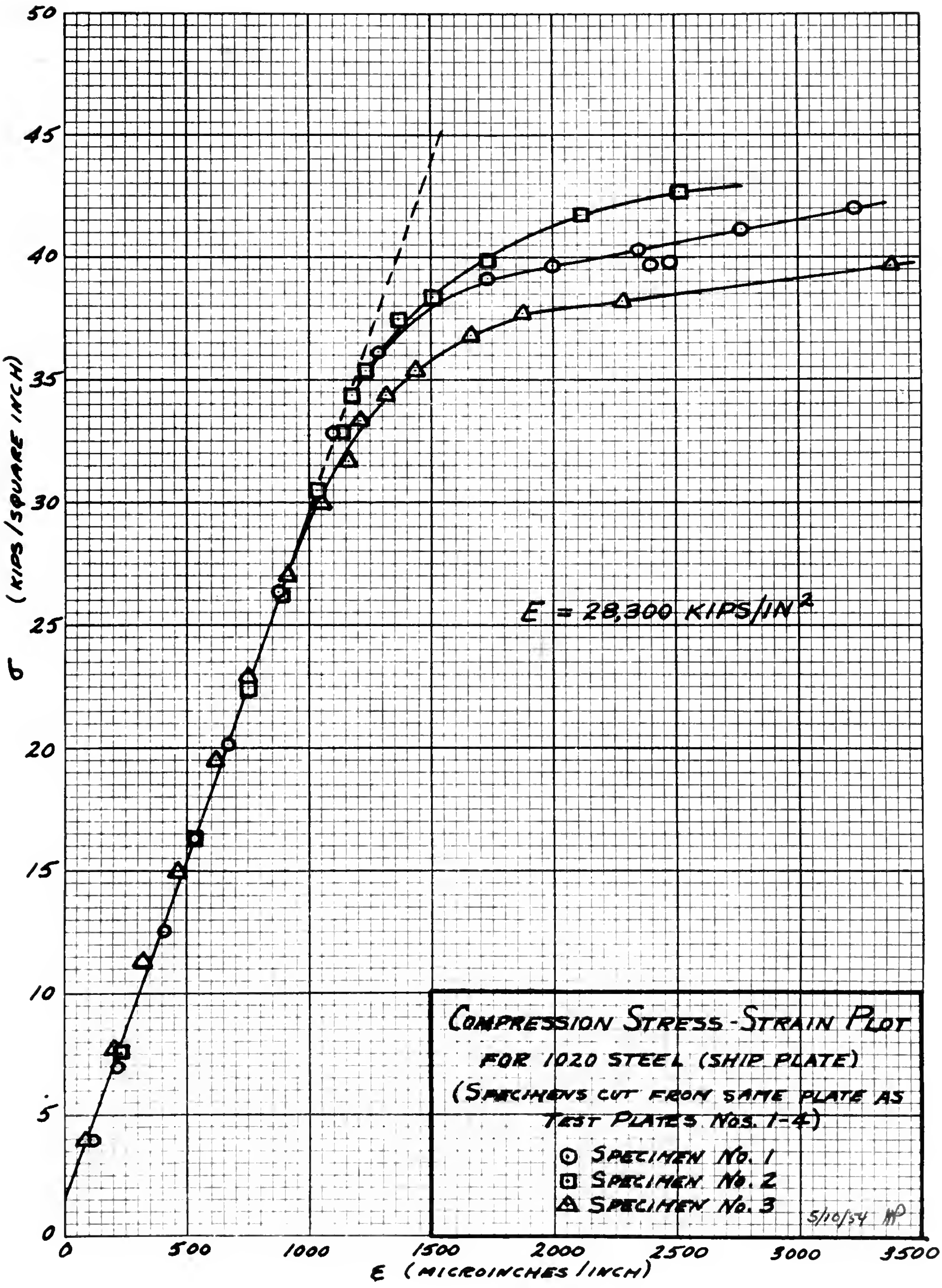
F. DETERMINATION OF YOUNG'S MODULUS

To determine the modulus of elasticity of the test specimens in compression, a compression jig using the single plate method designed and built by Pellegrene [10] was used.

Briefly the jig consists of a frame which holds the guide plates and specimen in place and contains a subpress in the form of a plunger which rests on the specimen. The guide plates are milled with triangular grooves which run the length of the specimen. The grooves opposite each other in the guide plates are staggered so that lateral extension of the specimen is permitted when a load is applied. Huggenberger strain gages of one inch base length are attached to opposite edges of the specimen through apertures in the frame. Guide plate pressure and centering is accomplished by adjusting four [illegible] knurled screws until they are just hand tight.

The specimens were rough cut from the same plate and in the same direction as were the test plates. These plates and the compression specimens received the same surface preparation such as [illegible] blasting or pickling.

Three specimens were prepared and tested in the jig [illegible]. The thickness was determined in the same manner as that used for the test plates. All specimens [illegible] 0.8125 ± 0.0005 which conform to the ASTM standards [illegible] tion. Compression Specimen). The results of the tests are shown in Fig. 17. There are several possible sources of error [illegible] here. The grooved guide plates do not completely [illegible] lateral extension. [illegible]



50
45
40
35
30
25
20
15
10
5
0
σ (KIPS/SQUARE INCH)
0
500
1000
1500
2000
2500
3000
3500
ε (MICROINCHES/INCH)
E = 28,300 KIPS/IN²
COMPRESSION STRESS-STRAIN PLOT
FOR 1020 STEEL (SHIP PLATE)
(SPECIMENS CUT FROM SAME PLATE AS
TEST PLATES NOS. 1-4)
SPECIMEN NO. 1
SPECIMEN NO. 2
SPECIMEN NO. 3
5/10/54 HP

yield point. There is a friction force developed between the specimen and its guide plates and also between the plunger and its guides, particularly if the loading should be uneven. Since the surface of these specimens was quite rough, care was taken not to tighten the knurled screws excessively especially since the prime result expected was the modulus rather than an accurate yield point. It should be noted however that such a friction force would tend to raise the value of E. Since the value of E found was slightly below that which one would normally expect for the type of steel tested, it is probable that the effect of such friction was not large and was reasonably constant for all tests.

The variations occuring in the stress-strain curve above the proportional limit are probably caused by uneveness in the load distribution across the loaded edge of the specimen. If one side was loaded more heavily than the other, the strain on that side would reach the proportioned limit before the other side and cause the curve to become more linear at various loads depending on the degree of uneveness. The points recorded above its proportioned limit were recorded after reasonable yielding had taken place but no attempt was made to wait until final values of the load and its strain were recorded. For this particular study an accurate value of the yield point is not necessary. It was felt that there was no need to pursue this phenomenon further. It is believed, however, in the future development of this test program, an accurate value of the yield stress will become important to the analysis of the data. Hence, the reasons for these variations should be pursued.

yield point. There [illegible] friction force resulting [illegible] the [illegible] and the guide plates and also between the [illegible] ticularly if the [illegible] should [illegible]. [illegible] specimens was [illegible], care was taken not to tighten the [illegible] screws excessively especially since the [illegible] was the modulus rather than an accurate yield point. [illegible] ever that such a friction force would [illegible]. Since the value [illegible] was slightly [illegible] normally expect for the type of steel tested, it is [illegible] effect of such friction was not large and was reasonably [illegible] for all tests.

The variations occurring in the stress-strain curves [illegible] proportional limit [illegible] probably caused by [illegible] tribution across the loaded ends of the specimen. [illegible] loaded more heavily than the others, [illegible] the proportional limit [illegible] come more [illegible]. The points [illegible] reasonably [illegible] until [illegible] particular [illegible]. It was felt [illegible]. It is believed, however, [illegible] an accurate value of the [illegible] analysis of the data. [illegible] be [illegible].

Specifications of Compression Test Procedure

Test Specimens:	Cut from same plate at test plates and in the same direction. Milled to 2.547 x 0.8435 inches. Three specimens tested.
Strain Gages:	Two Type A Huggenberger - 1 inch gage length No. 2195 Gage Factor 1051 No. 2201 Gage Factor 1041
Testing Machine:	<u>Testing Materials Lab. #203</u> Manufacturer - Riehle' Bros. Capacity - 20,000 pounds Specimens held in compression jig. Spherical head used.
Test Procedure:	Knurled bolts holding guide plates were tightened barely hand tight. Load of about 200 pounds applied and released several times to check zero readings on Huggenberger gages. Load applied in increments of approximately 500 pounds. Gages read at each step. Modulus of elasticity found by plotting stress vs. strain.

Specifications of Compression Test Procedure

Test Specimens: Cut from same plate as test plates and in the same direction. [illegible] inches. Three specimens tested.

Strain Gages: Two Type A Huggenberger - 1 inch gage lengths

No. 2195 Gage Factor [illegible]

No. 2201 Gage Factor [illegible]

Testing Machine: Testing Materials Lab. [illegible]

Manufacturer - [illegible] Bros.

Capacity - [illegible],000 pounds

Specimens held in [illegible]

Spherical head used.

Test Procedure: Knurled bolts holding gages [illegible] tightened barely hand tight. Load of about [illegible] pounds applied and released several times to check zero readings of Huggenberger gages. Load applied in increments of approximately 500 pounds. Gages read at each step. Modulus of elasticity found by plotting stress vs. strain.

APPENDIX G

ORIGINAL DATA

TABLE IV

Data Sheet

Date: 29 March 1954
Test No: 2
Plate No.: 1
Strain Indicator
Serial No. D58130
Type K
G.F. Setting 2.01

B-Edge Conditions

Load Application - Machine Jacks
Bar - 3/4" (Grooved)
Gasket - Neoprene

No Segments
No Shims

Load (kips)	Zero	2	4	6	8	10	12	14	Final Zero
Gage				Strain Indicator Readings (microinches/inch)					
1	8-608	592	589	589	582	574	568	564	604
2	7-177	155	158	152	150	145	141	136	177
3	9-249	210	205	200	197	190	187	180	252
4	4-407	401	395	385	376	363	357	343	400
5	8-752	738	738	725	721	711	708	707	754
6	7-737	730	728	720	718	710	708	691	741
7	6-782	788	792	783	770	749	721	691	783
8	5-557	535	531	538	542	552	570	582	562
9	7-179	191	184	171	152	141	128	107	188
10	8-866	821	810	800	795	790	791	792	870
11	6-651	666	664	662	662	659	660	657	657
12	6-1017	972	966	960	952	944	940	930	1020
13	8-873	850	847	841	840	836	836	833	880
14	9-267	261	250	240	230	220	208	193	270
15	7-423	380	380	376	372	367	365	361	428
16	7-907	1067	1077	1071	1069	1061	1058	1050	1050*
17	8-830	797	798	791	790	787	787	783	837
18	9-242	272	269	264	260	250	241	230	246
19	6-1000	970	962	957	953	948	950	947	1009
20	7-714	740	732	720	710	694	682	668	718
21	9-121	090	079	071	067	059	054	053	128
22	8-200	220	210	201	190	178	168	148	201
23	7-1060	1022	1013	1008	1002	996	992	992	1066
24	8-411	426	418	406	400	390	382	367	412
25	8-936	891	887	880	875	868	863	856	941
26	8-341	347	332	320	310	297	287	273	342
27	6-850	790	780	768	760	751	749	745	856
28	7-911	920	913	902	894	882	870	857	911
29	7-198	161	157	148	141	134	128	121	202
30	7-762	765	762	758	755	750	747	738	762
31	7-460	454	450	447	441	435	430	420	462
32	8-117	095	091	088	087	081	082	080	118
33	8-651	655	641	629	617	600	580	557	653
34	5-985	952	952	947	945	944	949	953	987
35	4-750	758	738	722	705	679	644	595	750
36	6-1030	1010	1026	1034	1047	1061	1090	1130	1040

TABLE IV

Data Sheet

B-Ridge Conditions

Date: 29 March 1954
Test No: 2
Plate No.: 1
Strain Indicator
Serial No. D58130
Type K
G.F. Setting 2.02

Load Application - Machine
Jacks
Bar - 3/4" (Grooved)
Gaskets - Neoprene

No Segments
No Fillets

Strain Indicator Readings (micro-inches/inch)

Gage \ Load (kips)	Zero	2	4	6	8	10	12	14	Final Zero
1	8-608	592	589	589	582	574	568	564	[illegible]
2	7-177	155	158	152	150	145	141	136	[illegible]
3	9-249	220	205	200	197	190	187	180	[illegible]
4	4-407	401	395	385	376	363	357	343	[illegible]
5	8-752	738	738	725	721	721	708	707	[illegible]
6	7-737	730	728	720	718	710	700	694	[illegible]
7	6-782	788	792	783	770	749	721	690	[illegible]
8	5-557	535	531	538	542	557	570	588	[illegible]
9	7-179	192	184	171	152	141	128	107	[illegible]
10	8-846	821	810	800	795	790	791	792	[illegible]
11	6-651	666	664	662	648	659	660	657	[illegible]
12	6-1017	972	966	960	952	944	940	930	[illegible]
13	8-873	850	847	841	840	838	836	833	[illegible]
14	9-267	261	250	240	230	220	200	193	[illegible]
15	7-423	380	380	376	372	367	365	361	[illegible]
16	7-907	1067	1077	1071	1069	1061	1058	1056	[illegible]
17	8-830	797	798	792	796	787	787	785	[illegible]
18	9-242	272	269	264	260	250	241	234	[illegible]
19	6-1000	970	962	957	953	948	950	947	[illegible]
20	7-714	740	732	720	710	694	682	666	[illegible]
21	9-121	090	079	071	067	059	054	053	[illegible]
22	8-200	220	210	201	190	175	165	[illegible]	[illegible]
23	7-1060	1022	1013	1008	1002	996	992	992	[illegible]
24	8-412	426	418	408	400	390	382	[illegible]	[illegible]
25	8-936	891	887	880	875	868	863	[illegible]	[illegible]
26	8-341	347	332	320	316	297	287	[illegible]	[illegible]
27	6-850	790	780	768	760	754	749	[illegible]	[illegible]
28	7-711	920	912	902	894	882	870	[illegible]	[illegible]
29	7-198	161	157	144	141	134	128	[illegible]	[illegible]
30	7-762	765	762	754	755	750	747	[illegible]	[illegible]
31	7-460	454	450	447	441	435	430	[illegible]	[illegible]
32	8-117	095	091	080	067	[illegible]	[illegible]	[illegible]	[illegible]
33	8-651	655	641	629	617	[illegible]	[illegible]	[illegible]	[illegible]
34	5-905	952	952	947	945	944	[illegible]	[illegible]	[illegible]
35	4-750	758	738	732	736	[illegible]	[illegible]	[illegible]	[illegible]
36	6-1093	1010	1084	1091	1047	1061	[illegible]	[illegible]	[illegible]

TABLE IV

Data Sheet

Date: 29 March 1954
Test No.: 2 (contd.)
Plate No.: 1
Strain Indicator
Serial No. 058130
Type K
G.F. Setting 2.01

B-Edge Conditions

Load Application - Machine Jacks
Bar - 3/4" (Grooved)
Gasket - Neoprene

No Segments
No Shims

Strain Indicator Readings (microinches/inch)

Load (kips) / Gage	Zero	2	4	6	8	10	12	14	Final Zero
37	6-521	510	524	534	548	557	573	570	525
38	7-479	461	448	428	407	380	348	334	480
39	8-463	457	458	460	466	470	472	475	466
40	8-457	439	429	413	400	361	365	351	459
41	8-1000	988	980	970	968	964	960	952	1007
42	8-952	937	923	910	900	891	882	880	958
43	8-258	241	232	218	202	182	162	143	263
44	7-557	563	561	564	571	578	587	591	560
45	8-173	151	142	130	119	103	090	071	180
46	7-719	740	747	747	750	751	758	760	725
47	8-748	724	711	698	683	667	649	630	755
48	7-046	076	083	088	092	096	100	105	051
49	7-410	380	367	349	330	310	290	269	416
50	7-248	272	276	280	288	296	308	316	252
51	7-540	521	513	504	498	487	475	462	548
52	4-820	838	841	842	843	842	844	848	826
53	6-390	367	358	329	312	295	278	255	398
54	7-377	376	460	438	370	-	336	335	352
55	7-828	794	770	740	710	674	639	604	838
56	8-223	243	252	259	271	286	302	318	228
57	8-978	920	906	890	876	858	840	820	983
58	6-410	430	436	438	444	450	459	465	411
59	7-1020	962	960	935	920	900	880	856	1028
60	8-477	507	511	517	521	524	528	530	480
61	7-554	514	503	488	470	453	438	419	561
62	5-670	686	682	680	682	682	684	688	675
63	8-661	637	638	630	625	621	623	620	667
64	8-333	341	330	323	317	305	289	282	339

Deflection Readings at 14 kips Load
(from a reference line)

Gage Position	Reading
1	0.3832-.0020
3	0.3915-.0015
5	0.2920-.0060

TABLE V

Data Sheet

Date: 31 March 1954
Test No.: 5
Plate No.: 1
Strain Indicator
    Serial No. D58130
    Type K
    G.F. Setting 2.01

B-Edge Conditions

Load Application - Machine Jacks
Bar - 3/4" (Flat)
Gasket - Single Solder 3/32" diam. (new)

No Segments
No Shims

Load (kips)	2	20	Zero	10	20	Gage	Zero	10	20
Pi	60	520	0	250	520		0	250	520
Gage	d (inches)		Strain Indicator Reading (microinches/inch)						
7	.315*	.325*	6-803	660	521	37	6-520	674	705
8			5-536	561	620	38	7-467	240	150
9	.318	.335	7-200	214	105	39	8-460	447	429
10			8-861	870	818	40	8-454	392	323
11	.322	.356	6-646	610	644	41	8-1002	977	979
12			6-1020	977	878	42	8-957	912	867
13	.341	.400	8-872	846	851	43	7-1262	1206	1207
14			9-269	249	187	44	7-560	559	508
15	.362	.435	-	-	-	45	7-1180	1139	1120
16			-	-	-	46	7-720	720	663
17	.360	.434	8-832	881	932	47	8-748	715	666
18			8-1240	1144	1016	48	6-1051	1014	980
19	.350	.420	6-1000	1051	1092	49	7-414	420	430
20			7-710	562	429	50	6-1249	1186	1142
21	.319	.375	-	-	-	51	7-544	498	490
22			-	-	-	52	4-815	791	749
23	.307	.318	7-1060	975	994	53	6-397	321	267
24			8-402	418	316	54	7-343	340	302
25	.292	.257	8-930	869	834	55	7-837	753	711
26			8-333	341	278	56	8-212	218	198
27	.267	.265	6-848	774	718	57	8-987	908	830
28			7-903	890	850	58	6-384	398	411
29	.237	.223	6-1198	1140	1110	59	7-1021	827	681
30			7-753	733	673	60	8-473	569	609
31	.211	.194	7-457	370	309	61	7-558	426	313
32			7-1112	1107	1089	62	5-666	733	776
33	.219	.198	8-650	586	504				
34			5-971	977	997				
35	.231*	.225*	4-773	495	152				
36			6-990	1180	1140				

* Readings taken between gage positions 7 & 9 or 33 & 35

Gage pairs 15, 16; 21,22; 63, 64; 65, 66 - no readings

TABLE VI

Data Sheet

Date: 11 April 1954 B-Edge Condition
Test No.: 12
Plate No.: 1 Load Application - Pump Segments
Strain Indicator Jacks Shims
Serial No. D58130 Bar - 1/8"
Type K Gasket: Double Solder 1/16" diam. ( )
G.F. Setting 2.01 Remarks: Plate reversed on a diagonal

Load 20 (kips)		Zero	2	10	20	Gage	Zero	2	10	20
Gage	d (in.)	Strain Indicator Reading (microinches/inch)								
7	.196*	6-793	767	789	774	37	6-502	504	540	597
8		5-540	553	533	534	38	7-464	452	399	334
9	.145	7-302	276	264	247	39	8-447	450	463	481
10		8-827	821	773	729	40	8-435	402	341	260
11	.096	6-641	636	648	670	41	8-990	991	997	1011
12		6-1022	1007	983	948	42	8-945	937	912	889
13	.089	8-861	848	851	872	43	8-242	230	230	238
14		9-264	245	190	114	44	7-561	549	497	440
15	.102	7-411	403	420	457	45	7-1163	1159	1160	1181
16		7-900	907	876	841	46	7-726	724	710	685
17	.137	8-815	795	797	819	47	8-730	719	699	696
18		8-1251	1238	1190	1119	48	6-1061	1046	1003	958
19	.172	6-990	978	990	1025	49	7-390	382	373	379
20		7-723	720	704	674	50	7-254	254	246	234
21	.210	8-1103	1088	1085	1098	51	7-520	490	455	430
22		7-582	578	531	465	52	4-816	820	793	756
23	.276	7-1041	1034	1040	1059	53	6-375	367	352	349
24		8-402	403	389	361	54	7-340	344	337	330
25	.298	8-916	896	887	879	55	7-809	796	757	718
26		8-334	325	289	236	56	7-1204	1191	1166	1141
27	.302	6-831	835	832	839	57	8-960	960	944	930
28		7-895	881	872	858	58	6-367	360	352	351
29	.266	6-1184	1185	1159	1135	59	7-994	988	948	902
30		7-744	720	700	670	60	8-453	439	418	395
31	.226	7-447	445	451	454	61	7-525	521	509	492
32		7-1099	1095	1080	1070	62	5-646	647	647	652
33	.212	8-630	611	645	644	63	8-643	620	581	530
34		5-963	959	875	815	64	8-309	305	286	277
35	.207*	4-683	669	716	740	65	7-705	711	699	682
36		6-1043	1054	1000	974	66	7-703	694	695	707

* Readings taken between gages 7 & 9 or 33 & 35

TABLE VII

Data Sheet

Date: 12 April 1954
Test No.: 14
Plate No.: 1
Strain Indicator
Serial No. D58130
Type K
G.F. Setting 2.01

B-Edge Condition

Load Application - Pump Jacks
Bar - 3/4" (Flat)
Gasket - Double Solder 1/16" diam. (new)
Remark: Repeatability Run

Segments
Shims

Strain Indicator Reading (microinches/inch)

Load (kips) / Gage	Zero	2	10	20	Gage	Zero	2	10	20	20 d (inches)
7	6-814	783	765	670	37	6-520	528	588	618	.193*
8	5-553	541	524	570	38	7-477	464	376	278	
9	7-315	303	283	240	39	8-462	470	467	469	.199
10	8-843	837	814	727	40	8-457	450	396	322	
11	6-650	649	648	639	41	8-1005	1012	986	972	.213
12	6-1024	1019	972	907	42	8-963	940	890	813	
13	8-871	873	862	877	43	8-1263	260	242	232	.257
14	9-278	260	180	090	44	7-578	557	520	455	
15	7-424	419	427	468	45	7-1179	1172	1160	1167	.286
16	7-908	891	844	760	46	7-743	719	681	626	
17	8-829	828	816	823	47	8-747	747	728	718	.283
18	8-1262	1260	1207	1110	48	6-1078	1071	1040	997	
19	6-1003	997	990	1019	49	7-409	396	362	340	.262
20	7-732	727	667	582	50	7-268	260	221	158	
21	8-1120	1110	1102	1118	51	7-539	531	491	459	.208
22	7-594	583	532	450	52	4-840	837	814	774	
23	7-1050	1050	1047	1064	53	6-390	368	312	249	.164
24	8-424	400	372	299	54	7-358	346	327	293	
25	8-928	913	883	868	55	7-826	791	720	637	.123
26	8-353	343	296	220	56	7-1224	1209	1178	1136	
27	6-848	857	810	777	57	8-975	967	919	862	.102
28	7-911	896	842	791	58	6-390	390	370	350	
29	6-1202	1205	1180	1150	59	7-1012	1010	968	922	.079
30	7-759	755	730	679	60	8-471	469	460	443	
31	7-468	470	440	417	61	7-544	540	508	469	.082
32	7-1110	1108	1076	1044	62	5-660	661	653	641	
33	8-658	658	661	640	63	8-663	640	569	492	.141
34	5-967	963	900	841	64	8-325	317	290	269	
35	4-701	702	822	804	65	7-732	698	692	682	.173*
36	6-1072	1022	869	820	66	7-715	719	682	632	

* Readings taken between gages 37 & 39 or 63 & 65

TABLE VIII

Data Sheet

Date: 14 April 1954
Test No.: 16
Plate No.: 1
Strain Indicator
Serial No. D58130
Type K
G.F. Setting 2.04

B-Edge Conditions

Load Application - Pump
No Jacks
Bar - 3/4" (flat)
Gasket - Double Solder
1/16" diam. (new)

Segments
Shims

Load (kips) Gage	Zero	2	10	20	Gage	Zero	2	10	20
7	6-827	820	835	814	37	6-582	616	664	672
8	5-520	515	418	362	38	7-371	325	231	145
9	7-298	280	277	248	39	8-433	441	442	438
10	8-787-	780	726	657	40	8-398	373	332	288
11	6-646	637	625	620	41	8-958	947	920	906
12	6-1002	992	952	892	42	8-905	884	826	747
13	8-833	832	825	830	43	7-1230	1223	1205	1198
14	8-1224	1217	1174	1119	44	7-538	521	477	413
15	7-404	409	398	408	45	7-1143	1141	1120	1121
16	7-870	861	798	725	46	7-700	680	641	597
17	8-788	791	800	812	47	8-701	690	670	660
18	8-1202	1187	1148	1080	48	6-1046	1033	998	948
19	6-988	981	970	971	49	7-381	372	347	338
20	7-697	691	619	533	50	7-234	230	203	171
21	8-1067	1064	1062	1071	51	7-507	496	451	406
22	7-559	567	527	461	52	4-842	835	811	787
23	7-1015	1015	1009	1007	53	6-370	352	308	248
24	8-383	391	360	301	54	7-333	331	310	278
25	8-867	864	838	792	55	7-777	761	708	625
26	8-320	311	265	170	56	7-1194	1190	1162	1134
27	6-813	807	778	745	57	8-910	899	852	807
28	7-888	890	852	804	58	6-384	377	366	360
29	6-1165	1163	1155	1144	59	7-964	955	910	858
30	7-734	738	720	703	60	8-436	433	422	419
31	7-430	424	392	361	61	7-504	486	430	370
32	7-1080	1079	1048	1010	62	5-662	670	665	675
33	8-601	617	588	562	63	8-610	542	467	350
34	5-968	945	872	812	64	7-1291	1281	1261	1260
35	4-711	800	842	807	65	7-678	580	486	326
36	6-1043	847	727	647	66	7-702	722	702	751

TABLE IX

Data Sheet

Date: 14 April 1954
Test No.: 16 (continued)
Plate No.: 1
Strain Indicator
Serial No. D58130
Type K
G.F. Setting 2.04

B-Edge Conditions

Load Application - Pump
No Jacks
Bar - 3/4" (flat)
Gasket - Double Solder
1/16" diam. (new)

Segments
Shims

Load Kips	Zero	2	20
Gage	Deflection - d(inches)		
37	.185*	.190*	.184*
39	.186	.193	.179
41	.189	.207	.181
43	.218	.246	.206
45	.247	.284	.235
47	.253	.291	.242
49	.252	.280	.235
51	.226	.239	.205
53	.219	.226	.195
55	.209	.188	.176
57	.190	.157	.160
59	.155	.133	.138
61	.136	.121	.125
63	.172	.150	.154

* Readings taken between gage positions 37 and 39.

TABLE X

Data Sheet

Date: 19 April 1954
Test No.: 19 (continued)
Plate No.: 1
Strain Indicator
Serial No. D58130
Type K
G.F. Setting 2.01

B-Edge Conditions

Load Application - Pump Jacks
Bar - 3/4" (flat)(1)
Gasket - Double Solder 1/16" diam.

Segments Shims

Load Kips	Zero	20		Zero	20
Gage	Strain		Gage	Strain	
7	6-851	1057	37	6-576	550
8	5-559	235	38	7-461	417
9	7-339	367	39	8-490	400
10	8-857	642	40	8-470	370
11	6-676	695	41	8-1028	961
12	6-1041	890	42	8-979	893
13	8-899	942	43	7-1288	1245
14	8-1293	1155	44	7-592	530
15	7-450	500	45	7-1204	1161
16	7-920	765	46	7-761	690
17	8-857	909	47	8-770	668
18	8-1269	1111	48	6-1089	990
19	6-1037	1030	49	7-433	305
20	7-733	563	50	6-1284	1200
21	8-1161	1130	51	7-558	475
22	7-582	466	52	4-860	802
23	7-1077	1024	53	6-404	341
24	8-450	371	54	7-388	350
25	8-943	833	55	7-836	712
26	8-381	301	56	7-1260	1209
27	6-862*	760	57	8-986	820
28	7-943	892	58	6-428	396
29	6-1217	1097	59	7-1027	900
30	7-790	750	60	8-500	500
31	7-484	387	61	--	--
32	7-1140	1118	62	--	--
33	8-672	647	63	--	--
34	5-1002	859	64	--	--
35	--	--	65	--	--
36	--	--	66	--	--

* Doubtful reading.

(1) Hump in lower bar over bottom jack and under gages 17 and 18.

TABLE I

Data Sheet

Date: 19 April 1951
Test No.: 19 (continued)
Plate No.: 1
Strain Indicator
Serial No. D58130
Type K
G.F. Setting 2.01

2-Edge Conditions

Load Application - Pump Jacks
Bar - 3/4" (Flat)(1)
Gasket - Double Solder 1/16" diam.

Segments Shims

Load Kips	Zero	20		Zero	20
Gage	Strain		Gage	Strain	
7	6-851	1057	37	6-976	550
8	5-559	235	38	7-461	417
9	7-339	367	39	8-490	400
10	8-857	642	40	8-470	370
11	6-676	695	41	8-1028	961
12	6-1041	890	42	8-979	893
13	8-899	942	43	7-1208	1245
14	8-1293	1155	44	7-592	530
15	7-450	500	45	7-1204	1161
16	7-920	765	46	7-761	690
17	8-857	909	47	8-770	848
18	8-1269	1111	48	6-1089	790
19	6-1037	1030	49	7-433	305
20	7-733	563	50	6-1284	1200
21	8-1161	1130	51	7-556	475
22	7-582	466	52	4-860	801
23	7-1077	1024	53	6-404	342
24	8-450	371	54	7-388	350
25	8-943	833	55	7-836	712
26	8-381	301	56	7-1240	1209
27	6-862*	760	57	8-984	880
28	7-943	892	58	6-420	390
29	6-1217	1097	59	7-1087	970
30	7-790	750	60	8-580	500
31	7-464	387	61	--	--
32	7-1140	1118	62	--	--
33	6-672	647	63	--	--
34	5-1002	259	64	--	--
35	--	--	65	--	--
36	--	--	66	--	--

* Doubtful reading.

(1) Bump in lower bar over between [illegible]

TABLE XI

Data Sheet

Date: 19 April 1954
Test No.: 19
Plate No.: 1
Strain Indicator
Serial No. D58130
Type K
G.F. Setting 2.01

B-Edge Conditions

Load Application - Pump Jacks
Bar - 3/4" (flat)(1)
Gasket - Double Solder 1/16" diam.

Segments
Shims

Gage	3	4	67	68
Load (kips)	Strain Indicator Reading (microinches/inch)			
Zero	8-1384	4-500	7-1328	7-635
2	1360	505	1331	602
4	1364	484	1350	566
6	1368	463	1364	535
8	1370	443	1380	503
10	1373	421	1394	470
12	1378	398	1412	439
14	1382	374	1431	403
16	1390	346	1452	363
18	1401	317	1478	323
20	1417	280	1509	276
22	1438	239	1541	227
24	1468	182	1581	166
26	1510	(120)	1641	(91)
		(3-1120)		(6-1091)
28	1580	1032	1706	1005
30	1678	910	1810	883
32	(1912)	617	(1940)	670
	(9-930)		(8-980)	
34	1220	318	1269	348
36	1995	2-320	1750	5-720

Maximum Load - 36.2 kips

(1) Bump in lower bar over bottom jack and under gages 17 and 18.

TABLE XII

Data Sheet

Date: 19 April 1954
Test No.: 20
Plate No.: 2
Strain Indicator
Serial No. D58130
Type K
G. F. Setting 2.01

B-Edge Conditions

Load Application - Pump Jacks
Bar - 3/4" (Flat)
Gasket - Double Solder 1/16" diam.

Segments
Shims

Gage	11	22	3	4*	5	6	7	8	9	10
Load Kips	Strain Indicator Reading (microinches/inch)									
Zero	5-1202	7-848	5-1261	A4-935	8-430	5-994	7-1017	7-564	5-911	5-1495
2	1182	841	1210	930	399	1010	988	574	882	1506
4	1165	845	1182	940	373	1024	955	586	844	1518
6	1143	847	1149	952	342	1044	920	602	828	1530
8	1122	850	1117	962	303	1064	880	622	801	1546
10	1093	854	1079	982	262	1097	830	648	767	1563
12	1067	867	1032	1011	211	1133	770	681	730	1588
14	1031	880	982	1057	152	1180	702	724	682	1619
16	994	902	919	1082	7-1090	1232	626	774	633	1649
18	947	927	850	1141	1003	1304	525	848	570	1702
20	888	963	753	1219	889	1402	400	936	494	1760
22	800	1020	600	1347	698	1566	167	1110	357	1872
24	643	1126	337	1117	393	1799	6-880	1302	211	1973
226	438	1294	4-960	1770	6-890	7-212	311	1720	4-870	6-1249
28	4-1795	1763	000	A5-1487	5-800	1050	5-390	8-1310	450	1540

Maximum Load - 29.5 Kips

* Erratic readings

TABLE XIX

Data Sheet

E-Edge Conditions

Date: 19 April 1954
Test No.: 20
Plate No.: 2
Strain Indicator
Serial No. D58130
Type K
G. F. Setting 2.01

Load Application - Pump Jacks
Bar - 3/4" (flat)
Gasket - Duroid Solder 1/16" diam.

Segments Skins

Gage	11	22	3	4	5	6	7	8	9	10
Load Kips	Strain Indicator Reading (microinches/inch)									
Zero	1202	848	1261	955	830	954	1017	524	911	1495
2	1182	844	1210	930	399	1010	988	574	602	1514
4	1165	845	1182	940	373	1024	955	586	584	1518
6	1143	847	1149	952	342	1044	920	602	628	1530
8	1122	850	1127	962	305	1064	880	622	601	1546
10	1093	854	1079	982	262	1097	839	648	727	1563
12	1057	867	1032	1011	211	1133	770	681	730	1588
14	1031	880	982	1057	152	1180	707	724	682	1619
16	994	902	919	1082	1090	1232	626	774	633	1649
18	947	927	850	1141	1003	1304	525	848	570	1702
20	888	963	753	1219	889	1402	400	936	484	1760
22	800	1020	600	1347	698	1546	187	1110	377	1878
24	643	1126	337	1117	397	1779	280	1302	211	1973
26	438	1294	360	2770	590	212	211	1720	370	1829
28	1795	1763	000	1187	800	1050	390	1310	150	1540

Maximum Load - 29.5 Kips

* Erratic readings

TABLE XIII

Data Sheet

Date: 19 April 1954
Test No.: 20 (continued)
Plate No.: 2
Strain Indicator
Serial No. D58130
Type K
G. F. Setting 2.01

B-Edge Conditions

Load Application - Pump Jacks

Segments Shims

Gasket - Double Solder 1/16" diam.

Gage / Load Kips	1	3	5	7	9
	Deflection - d(inches)				
1	.218	.276	.296	.265	.289
10	.205	.288	.302	.294	.314
15	.225	.322	.371	.317	.342
20	.257	.373	.429	.404	.384
24	.309	.468	.522*	.522*	.457

* Limit of dial gage.

TABLE XIV

Data Sheet

Date: 21 April 1954
Test No.: 21
Plate No.: 3
Strain Indicator
Serial No. D58130
Type K
G.F. Setting 2.01

Load Application - Pump Jacks
Bar - 3/4" (G)
Gasket - Single 3/32" solder

Segments
Shims

Load Kips	Gage No. 1	Gage No. 2	-24"	-18"	-12"	-6"	cl	6"	12"	18"	24"
	Strain Reading (μin./in.)		Deflection (d') (in. x 1000)								
Zero	8-542	8-769	65	86	85	82	64	60	40	30	05
2	534	762	57	89	83	73	50	43	26	17	03
4	521	760	55	84	76	67	42	33	15	9	-05
6	507	760	55	82	75	65	36	12	14	9	-02
8	489	761	55	81	76	66	37	30	12	8	--
10	465	766	55	83	75	64	37	29	10	6	--
12	437	771	55	82	74	61	34	24	8	4	--
14	421	784	54	78	70	59	32	21	3	2	--
			x385 (Reset dial gage)								
16	398	794	x384	405	396	384	357	343	325	324	317
18	365	810	383	403	391	356	350	337	320	320	314
20	327	831	382	400	383	372	342	328	311	313	310
22	271	868	380	394	378	360	330	316	300	306	307
24	200	921	376	387	366	346	314	299	287	295	302
	7-1200										
26	1082	1018	368	370	346	322	289	275	267	281	295
28	874	1192	356	340	305	275	235	222	224	252	282
30	057	1871	299	227	166	133	106	112	129	180	252

Position from Vertical Centerline (on Horizontal Centerline)

Maximum Load - 32,500 pounds

TABLE XV

Data Sheet

Date: 21 April 1954
Test No.: 22
Plate No.: 4
Strain Indicator
Serial No. D58130
Type K
G.F. Setting 2.01

Load Application - Pump Jacks
Bar - 3/4" (G)
Gasket - Single 3/32" diameter solder

Segments
Shims

Load Kips	Gage No. 1	Gage No. 2	-24"	-18"	-12"	-6"	©L	6"	12"	18"	24"
	Strain Reading (μin./in.)		Deflection (d') (in. x 1000)								
Zero	8-1307	7-1142	266	225	171	120	102	107	154	168	241
2	1281	1147	256	215	160	131	94	95	141	167	238
4	1259	1156	252	203	152	105	88	86	134	159	235
6	1225	1171	252	200	146	105	80	84	131	159	235
8	1193	1188	247	195	140	99	72	76	128	152	235
10	1157	1210	247	188	129	86	65	70	124	153	235
12	1111	1234	242	180	120	83	56	60	114	147	231
14	1059	1269	236	171	106	70	42	55	109	144	229
16	1000	1302	230	160	92	50	31	40	98	138	225
18	930	1361	226	148	73	27	18	30	90	131	222
20	851	1416	217	130	56	12	0	8	72	119	218
22	741	1510	206	111	10	--	--	--	58	105	214
24	598	1615									
26	410	1772									
28	042	1965									
30	7-250	8-1640									

Maximum Load - 33,500 pounds

TABLE XVI

Initial Unfairness of Specimens

Location from vertical centerline (on horizontal centerline)	Plate No. 1	2	3	4
	Initial deflection d_o (in.)			
-24"	---	.055	.004	---
-18"	---	.036	-.002	.066
-15"	---	.043	---	---
-12"	---	.053	-.007	.074
- 9"	.085*	.057	---	---
- 6"	---	.057	-.009	.057
- 3"	---	.051	---	---
Centerline	---	.041	-.008	.071
+ 6	---	.037	.002	.066
+12	---	.031	.000	.033
+18	---	.041	.005	.020
+24	---	.054	.006	.021

* Maximum deflection under gage numbers 17 and 18, 8.04" from vertical centerline, 1.32 above horizontal centerline.

H. LITERATURE CITATIONS

1. Ballard, J. A., Lemmon, R. C., "An Investigation of the effects of Shear on the Compressive Strength of Plating", MIT Thesis, Course XIIIA, 1949.
2. Batdorf, S. B., Stein, M., "Critical Combinations of Shear and Direct Stress for Simply Supported Rectangular Flat Plates", NACA, TN-1223, 1947.
3. Batdorf, S. B., "Theories of Plastic Buckling", Jour. Aero. Sci., Vol. 16 #7, p. 405, July 1949.
4. Bengston, H. W., "Ship Plating Under Compression and Hydrostatic Pressure", Trans. SNAME, Vol. 47, p. 80, 1939.
5. Bijlaard, P. P., "Theory of the Plastic Stability of Thin Plates", Int. Ass'n of Bridge and Structural Engineers, Vol. 6, p. 45, 1940-41.
6. Bijlaard, P. P., "Theory and Tests on the Plastic Stability of Plates and Shells", Jour. Aero. Sci., Vol. 17, No. 9, p. 529, 1949.
7. Bleich, F. B., Ramsey, L. B., "A Design Manual on the Buckling Strength of Metal Structures", T and R Bull. No. 2-2, SNAME, 1951.
8. Bleich, F. B., <u>Buckling Strength of Metal Structures</u>, McGraw-Hill Book Co., 1952.
9. Budiansky, B., Hu, Pai C., "The Lagrangian Multiplier Method of Finding Upper and Power Limits to Critical Stresses of Clamped Plates", NACA, TN-1103, 1946.
10. Budiansky, B., Connor, R. W., "Buckling Stresses of Clamped Rectangular Flat Plates in Shear", NACA, TN-1559, 1948.
11. Budiansky, B., Connor, R. W., Stein, M., "Buckling in Shear of Continuous Flat Plates", NACA, TN-1565, 1948.

H. LITERATURE CITATIONS

1. Ballard, J. A., Lennon, R. C., "An Investigation of the effects of Shear on the Compressive Strength of Plating", MIT Thesis, Course XIIIA, 1949.
2. Batdorf, S. B., Stein, M., "Critical Combinations of Shear and Direct Stress for Simply Supported Rectangular Flat Plates", NACA, TN-1223, 1947.
3. Batdorf, S. B., "Theories of Plastic Buckling", Jour. Aero. Sci., Vol. 16 #7, p. 405, July 1949.
4. Bengston, H. W., "Ship Plating Under Compression and Hydrostatic Pressure", Trans. SNAME, Vol. 47, p. 80, 1939.
5. Bijlaard, P. P., "Theory of the Plastic Stability of Thin Plates", Int. Ass'n of Bridge and Structural Engineers, Vol. 6, p. 45, 1940-41.
6. Bijlaard, P. P., "Theory and Tests on the Plastic Stability of Plates and Shells", Jour. Aero. Sci., Vol. 16, No. 9, p. 529, 1949.
7. Bleich, F. B., Ramsey, L. B., "A Design Manual on the Buckling Strength of Metal Structures", 1 and 2 Bull. No. 2-2, SNAME, 1951.
8. Bleich, F. B., Buckling Strength of Metal Structures, McGraw-Hill Book Co., 1952.
9. Budiansky, B., Hu, Pai C., "The Lagrangian Multiplier Method of Finding Upper and Lower Limits to Critical Stresses of Clamped Plates", NACA, TN-1103, 1946.
10. Budiansky, B., Connor, R. W., "Buckling Stresses of Clamped Rectangular Flat Plates in Shear", NACA, TN-1559, 1948.
11. Budiansky, B., Connor, R. W., Stein, M., "Buckling in Shear of Continuous Flat Plates", NACA, TN-1565, 1948.

12. Capozzoli, L. J., "Buckling of Compressed Rectangular Plates With Clamped Intermittent Edge Supports", MIT Course I Thesis, 1950.

13. Cox, H. L., "Buckling of Thin Plates in Compression", British A.R.C., R and M No. 1554, 1933.

14. Cox, H. L., Clenshaw, W. J., "The Strength of Ships' Plating", Institute of Engineers and Shipbuilders, Vol. 81, p. 17, 1937-38.

15. Cox, H. L., "The Buckling of a Flat Rectangular Plate Under Axial Compression and Its Behavior After Buckling", British A.R.C., R and M No. 2041, May, 1945.

16. Cox, H. L., "The Buckling of a Flat Plate Under Axial Compression and Its Behavior After Buckling", British A.R.C., R and M No. 2175, Oct., 1945.

17. Cox, H. L., "Note on the Southwell Method for Estimating Critical Loads", British A.R.C., R and M No. 2696, Feb., 1947.

18. Donnell, L. H., "On the Application of Southwell's Method for the Analysis of Buckling Tests", S. Timoshenko 60th Anniversary Volume, p. 27, 1938.

19. Frankland, J. M., "The Strength of Ship Plating Under Edge Compression", David Taylor Model Basin Report 469, 1940.

20. Gerard, G., "Secant Modulus Method for Determining Plate Instability Above the Proportional Limit", Jour. Aero. Sci., Vol. 13, p. 38, 1946.

21. Handelman, G. H., Prager, W., "Plastic Buckling of a Rectangular Plate Under Edge Thrusts", NACA, TN-1530, TR-946, 1948.

22. Heimerl, G. J., "Determination of Plate Compressive Strengths", NACA, TN-1480, 1947.

23. Hill, H. N., "Chart for Critical Compressive Stress of Flat Rectangular Plates", NACA, TN-773, 1940.

12. Capozzoli, L. J., "Buckling of Compressed Rectangular Plates With Clamped Intermittent Edge Supports", MIT Course I Thesis, 1959.

13. Cox, H. L., "Buckling of Thin Plates in Compression", British A.R.C. R and M No. 1554, 1933.

14. Cox, H. L., Clenshaw, W. J., "The Strength of Ships' Plating", Institute of Engineers and Shipbuilders, Vol. 82, p. 171, 1938-39.

15. Cox, H. L., "The Buckling of a Flat Rectangular Plate Under Axial Compression and Its Behavior After Buckling", British A.R.C. R and M No. 2041, May, 1945.

16. Cox, H. L., "The Buckling of a Flat Plate Under Axial Compression and Its Behavior After Buckling", British A.R.C., R and M No. 2175, Oct., 1944.

17. Cox, H. L., "Note on the Southwell Method for Estimating Critical Loads", British A.R.C., R and M No. [illegible], Feb., 1944.

18. Donnell, L. H., "On the Application of Southwell's Method for the Analysis of Buckling Tests", S. Timoshenko 60th Anniversary Volume, p. 27, 1938.

19. Frankland, J. M., "The Strength of Ship Plating Under Edge Compression", David Taylor Model Basin Report 469, 1940.

20. Gerard, G., "Secant Modulus Method for Determining Plate Instability Above the Proportional Limit", Jour. Aero. Sci., Vol. 13, p. 38, 1946.

21. Handelman, G. H., Prager, W., "Plastic Buckling of a Rectangular Plate Under Edge Thrusts", NACA, TN 1530, TR-946, 1948.

22. Heimerl, G. J., "Determination of Plate Compressive Strengths", NACA TN-1480, 1947.

23. Hill, H. N., "Chart for Critical Compressive Stress of Flat Rectangular Plates", NACA, TN-773, 1940.

24. Hoff, N. J., "Note on Inelastic Buckling", Jour. Aero. Sciences, Vol. 2, No. 2, p. 163, 1944.

25. Hopkins, K. G., Rao, B. V. S. C., "The Initial Buckling of Flat Rectangular Panels Under Combined Shear and Compression", British A.R.C., R and M No. 1965, 1943.

26. Hu, Pai C., Lundquist, E. E., Batdorf, S. B., "Effect of Small Deviations from Flatness on Effective Width and Buckling of Plates in Compression", NACA, TN-1124, 1946.

27. Ilyushin, A. A., "The Elasto-Plastic Stability of Plates", N.A.C.A., T.M. 1188, 1946.

28. Ilyushin, A. A., "Stability of Plates and Shells Beyond the Proportional Limit", N.A.C.A., T.M. 1116, Oct. 1947.

29. Kollbrunner, C. F., "Das Ausbuelen der auf einseitigen, gleichmaesig verteillen Druck beanspruchten Platten in elastischen und plastischen Beriech", Zurich E.T.H. Inst. f. Baustatic No. 17, 1946.

30. Lahde, R., Wagner, H., "Experimental Studies of the Effective Width of Buckled Sheets", N.A.C.A. T.M. 814 (Transl), 1936.

31. Levy, S., "The Bending of Rectangular Plates with Large Deflections", N.A.C.A. T.M. 846, 1942.

32. Lundquist, E., Stowell, E., "Critical Compressive Stress for Flat Rectangular Plates Supported Along All Edges and Elastically Restrained Against Rotation Along the Unloaded Edges", N.A.C.A. T.R. 733, 1942.

33. Marguerre, K., "The Apparent Width of the Plate in Compression", N.A.C.A. T.M. 833 (Transl) 1937.

34. Maulbetsch, J. L., "Buckling of Compressed Rectangular Plates with Built-in Edges", Jour. Appl. Mech., Vol. 4, p. A-54, June, 1937.

35. Mitchell, E. H., "The Strength of Ships' Structure Under Compressive Stresses", Trans., I.N.A., Vol. 78, p. 21, 1936.

36. Montgomerie, J., "Experiments on the Compression of Samples of Deck Plating and the Application of the Results in Relation to the Thickness of Such Plating in Certain Conditions of Loading", Shipbuilder and Marine Engine Builder, Vol. 41, p. 213, April, 1934.

37. Muckle, W., "Resistance to Buckling of Light Alloy Plates", N.E. Coast Inst. Eng. and Shipb. Trans., Vol. 64, p. 223, March, 1948.

38. Peters, R. W., "Buckling Tests of Flat Rectangular Plates Under Combined Shear and Longitudinal Compression", N.A.C.A. T.N. 1750, 1948.

39. Pfingstag, F. W., Palmer, C. J., "Effect of End Fixity on the Strength of Plating in Compression", M.I.T. Course XIIIA Thesis, 1937.

40. Polychrone, D. A., "On The Buckling of Intermittently Supported Plates", M.I.T. Thesis, Course I, 1949.

41. Pride, R. A., Heimerl, G. J., "Plastic Buckling of Simply Supported Compressed Plates", N.A.C.A. T.N. 1817, 1949.

42. Rossell, H. E., "Strength of Plating in Compression", S.N.A.M.E., Trans., Vol. 43, p. 121, 1935.

43. Schnadel, G., "Knickung von Shiffsplatten", Werft Reederei Hafen, p. 461, 15 November, 1930.

44. Schnadel, G., "Strength of Transversely Stiffened Decks of Ships", Geo. Reissner Anniv. Vol., p. 256, 1949.

45. Schultz, F. E., Weschler, C. J., "Strength of Flat Plating Under Edge Compression", M.I.T. Thesis Course XIII, 1937.

46. Schuman, L., Back, G., "Strength of Rectangular Flat Plates Under Edge Compression", N.A.C.A. T.R. 356, 1930.

47. Sechler, E. E., "The Ultimate Strength of Thin Flat Sheets in Compression", Guggenheim Aeronautical Laboratory Pub. 27, California Institute of Technology, 1933.

48. Southwell, R. V., "On the Analysis of Experimental Observations in Problems of Elastic Stability", Proc. Roy. Soc. London, Ser. A, Vol. 135, p. 601, 1932.

49. Stowell, E. Z., Schwartz, E. B., "Critical Stress For an Infinitely Long Flat Plate with Elastic Restrained Edges Under Combined Shear and Direct Stress", N.A.C.A. W.R. L-340 (ARR - 3K13), 1943.

50. Stowell, E. Z., "Critical Shear Stress of an Infinitely Long Flat Plate with Equal Elastic Restraint Against Rotation Along Parallel Edges", N.A.C.A. W.R. L-476, (ARR - 3K12), 1943.

51. Stowell, E. Z., "A Unified Theory of Plastic Buckling of Columns and Plates", N.A.C.A. T.R. 898, 1948.

52. Stowell, E. Z., "Critical Shear Stress of an Infinitely Long Plate in the Plastic Region", N.A.C.A. T.N. 1681, 1948.

53. Stowell, E. Z., "Buckling Strength for Flat Plates and Sections", A.S.C.E., 1951.

54. Sweeney, R. J., "The Strength of Hull Plating Under Compression", U. S. Experimental Model Basin, Progress Reports 1 and 2, 1933.

55. Timoshenko, S., Theory of Elastic Stability, McGraw-Hill Book Co., 1936.

56. von Karman, T., Sechler, E. E., Donnell, L. H., "The Strength of Thin Plates in Compression", Transactions A. S. M. E., Vol. 54, APM-54-5-53, 1932.

57. Wang, Chi-Teh, "Non-linear Large Deflection Boundary Value Problems of Rectangular Plates", N.A.C.A. T.N. 1425, 1948.

58. Warren, S. F., "The Buckling of Thin Plates Subjected to Combined Loads", M.I.T. Thesis, Course I, 1947.

59. Yoshiki, M., "On the Buckling and Collapsing Strength of Ship's Hull Parts Under Compressive Load", Jap. Sci.,Rev., Vol. 1, p. 148, 1949-50.

60. Yoshiki, M. "A New Method to Determine the Critical Buckling Points of Rectangular Plates in Compression", International Congress of Applied Mechanics, Istanbul, 1952.

61. Yoshiki, M., Handbook of Elastic Stability, Corona Pub. Co., Tokyo, 1951.

58. Warren, S. F., "The Buckling of Thin Plates Subjected to Combined Loads", M.I.T. Thesis, Course I, 1947.

59. Yoshiki, M., "On the Buckling and Collapsing Strength of Ship's Hull Parts Under Compressive Load", Jap. Sci. Rev., Vol. 1, p. 143, 1949-50.

60. Yoshiki, M. "A New Method to Determine the Critical Buckling Points of Rectangular Plates in Compression", International Congress of Applied Mechanics, Istanbul, 1952.

61. Yoshiki, M., Handbook of Elastic Stability, Corona Pub. Co., Tokyo, 1951.